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HEAT EXCHANGER OPTIMIZATION.(U)

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Monterey, California

THESIS

HEAT EXCHANGER OPTIMIZATION

by

Conrad P. Hedderich

September 1980

Thesis Co-Advisors: M. Kelleher
G. Vanderplaats

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A general iteration free approximation method was used for the analysis which calculates the mean overall heat transfer coefficient and the overall pressure drop for many flow

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The code is not limited to surfaces found in the literature, but will accommodate any triangular pitch bank of finned tubes in multiple-pass configurations.

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HEAT EXCHANGER OPTIMIZATION

by

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Submitted in partial fulfillment of the
requirements for the degree of

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ABSTRACT

A computer code was developed for the analysis of air-cooled heat exchangers and was coupled with a constrained function minimization code to produce an automated air-cooled heat exchanger design and optimization program with many new capabilities.

A general iteration free approximation method was used for the analysis which calculates the mean overall heat transfer coefficient and the overall pressure drop for many flow arrangements, taking into account the variation of the heat transfer coefficients and the pressure drop with temperature and/or length of flow path.

The code is not limited to surfaces found in the literature, but will accommodate any triangular pitch bank of finned tubes in multiple-pass configurations.

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NOMENCLATURE

English Letter Symbols

A - total heat transfer area, in^2

c_p - specific heat, BTU/lbm-°F

$\dot{C}$ - heat capacity rate, BTU/hr-°F = $\dot{m}\, c_p$

D - diameter, in.

f - friction factor

F - LMTD correction factor

g_c - acceleration of gravity, 32.2 ft/sec^2

H - corrected heat transfer coefficient, BTU/hr-ft^2-°F

h - bank height, in.

J - Colburn factor

k - thermal conductivity, BTU/hr-ft-°F

L - length, in.

ℓ - fin height, in.

$\dot{m}$ - mass flow rate, lbm/hr

m - $\sqrt{2H_o/k_f t}$, ft^{-1}

n - number of ____ (used with appropriate subscript)

N - number of tubes

p - pressure, psi

P - pitch, in.

Pr - Prandtl number

$\dot{Q}$ - heat transfer rate, Btu/hr

r - radius, in.

R - heat transfer resistance, hr-ft^2-°F/BTU

Re - Reynolds number

s - distance between adjacent fins, in.

S - fin spacing center-to-center, in.

t - fin thickness, in.

T - true temperature, deg.

U - overall heat transfer coefficient, BTU/hr-ft^2-°F

w - bank width, in.

Greek Letter Symbols

ΔT - temperature difference, deg.

η - surface efficiency

ϕ - fin efficiency

μ - viscosity, lbm/ft-hr

ρ - density, lbm/ft^3

ψ - temperature correction

Subscripts

a - air

c - cold

f - fin

ff - free face

h - hot

i - inside

j - reference number, I or II

ℓ - limiting

L - longitudinal

m - mean

o - outside

p - pass

r - rows

t - transverse

T - total

w - water

x - cross-sectional (flow)

1 - entering

2 - leaving

∞ - ambient

Superscripts

$'$ - uncorrected

* - estimated

i - initial

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I. INTRODUCTION

A. BACKGROUND

The cooling of fluids by passing ambient air over extended tube surfaces is a relatively recent development in heat exchangers. Its application has come about cautiously, due to the usual reluctance to change from well established and well documented methods, i.e. the shell-and-tube heat exchanger.

However, concern for the environment and economic pressures have necessitated the use of air as a coolant. Smith [1] has listed some typical advantages of direct cooling with air as compared to cooling with water in a shell-and-tube exchanger:

a. *Eliminates the problem of temperature rise in, and* pollution of, water resources.

b. Enables plant location to be independent of a water supply.

c. Eliminates the necessity of much coolant piping.

d. Reduces heat exchanger maintenance costs by eliminating the need of descaling water-side surfaces. The mechanical drives will operate in a noncorrosive atmosphere.

e. Eliminates water treatment.

f. Limiting coolant temperatures is unnecessary.

g. Enables installation of exchangers at elevations above other operating equipment at no penalty, thus reducing ground area requirements.

With air colling becoming more and more competitive with water; even when water supplies are plentiful, an automated air-cooled heat exchanger design package (which could be used

for trade-off studies, first cut analysis, and conceptual design), would be of great use.

The design of an air-cooled cross flow heat exchanger is a complex task requiring the examination and optimization of a wide variety of heat transfer surfaces. Studies have shown that a poor choice of either the heat transfer surfaces or design parameters can more than double the costs charageable to a heat exchanger [2].

For the optimized design of heat exchangers with the computer; reliable, but fast, calculation methods for the mean overall heat transfer coefficient and the overall pressure drops are needed for the following reasons:

a. Conventional simple methods using mean values of temperatures as reference temperatures can lead to undesirable errors [3].

b. Numerical stepwise integrations are prohibitively time consuming.

B. REVIEW

A number of heat exchanger design methods have been proposed to determine the optimum heat exchanger design. Bergles, et al. [4], performed an evaluation of different objective functions for compact heat exchangers with different heat transfer surfaces, but the same specifications. The method did not include any actual optimization techniques, but results did show that a great improvement in heat exchanger performance can be made by proper selection of design parameters.

The method of Fax and Mills [5], used Lagrange multipliers to optimize a heat exchanger design under specified constraints. This technique required that the objective function and constraints be expressed explicitly and be differentiable throughout the range of interest. The total number of constraints had to be less than the total number of variables, and all constraints had to be equality constraints. The method was obviously restricted to a very limited number of problems.

For a unit section of an air-cooled exchanger of standard length and width equipped with specified fans, both Schoonman [6], and Joyce [7], used factorial searches of fin spacing, number of rows and air rate to maximize the ratio of heat transfer to cost. Because the number of sections must be rounded to an integer, this method only appears suitable for large exchangers.

Nakayama [3], used a similar approach requiring the plotting of heat transfer coefficients and pressure drops which presumably could be programmed into a factorial search.

Kern [8] derived an analytic expression for the annual cost as a function of air rate and number of rows. (This was done by using a constant tubeside heat transfer coefficient and an arithmetic mean temperature difference.) The optimum was found by setting partial derivatives equal to zero. Alternatively, if the airside is assumed to control, the optimum allocation of the components of total cost might be

found through geometric programming, as illustrated by Auriel and Wilde [9]. Oshwald and Kochenberger [10], also presented a geometric programming method for heat exchanger optimization and used it to select heat exchanger fluids considering power requirements, cost, tube diameters, velocities, temperature, and other physical properties.

After discussing factorial, univariate, and random search methods for an optimum design of a shell-and-tube exchanger, Briggs and Evans [11, 12], discuss a "logical search method"; or what Peters and Nicole [13], call "heuristics". With this method, an engineer makes use of selecting design variables close to the optimum to obtain an optimum design. The heuristic method is less scientific, and is useful only when computer time and storage are at a premium. Due to the large number of discrete variables encountered in air-cooled heat exchanger design, Peters and Nicole [13], chose to base their cost-optimizing design programs on heuristic algorithms (starting close to optimum), specific to the equipment under consideration.

Mott, et al. [2], discuss a computerized procedure for designing a minimum cost heat exchanger. The method minimizes a cost index expressed as a function of fluid pumping power. The algorithm imposed no constraints.

To this point, no mention has been made of applying the concepts and techniques of nonlinear programming to optimizing the design of heat exchangers. However, Palen, et al. [14], in 1974 proposed using the Complex Method [15], for the heat

exchanger optimization problem. They found a minimum cost shell-and-tube exchanger by varying six geometrical parameters. The Complex Method requires several feasible starting designs before optimization can be performed.

Johnson, et al. [16], coupled an existing shell-and-tube condenser design code with a constrained function minimization code to produce an automated marine condenser design program of vastly different complexity.

The most complete work to date has been accomplished by Afimiwala [17]. He has applied various nonlinear programming methods of optimization to the heat exchanger design problem; including an experimental interactive graphical approach and exterior penalty function techniques. The gradient based search methods of Davidson - Fletcher - Powell and conjugate gradient were used for the resulting unconstrained minimizations. The exterior penalty method is extremely useful, since an initial solution satisfying the constraints is not required. The gradient based search methods are efficient when considering computer time.

Finally, Fontein and Wassink [18] utilized the complex method of Nelder and Mead [19], and a steepest descent method [20], for optimizing a shell-and-tube exchanger.

It can be seen that although there are many methods that have been presented for heat exchanger optimization, each of the methods has its own limitations; none is completely general. Of all the design procedures cited above (those of which are applicable to cross flow air-cooled heat exchangers), all are

limited to the 120 individual surfaces found in the open literature [21] for the calculation of the air-side heat transfer coefficient and friction factor. Therefore, the designer is faced with choosing an optimum surface from a number of individual optimal designs calculated from one of the above methods. In addition, the above methods treat the overall heat transfer coefficient as a constant, or they become involved with time-consuming numerical stepwise integrations in an attempt to account for the varying heat transfer coefficients.

This paper tries to bridge this gap by presenting an optimization routine that: selects an optimal surface, takes into account the varying heat transfer coefficients and friction factors across the exchanger, performs each analysis in an iterative-free manner, and may start with an infeasible design.

C. METHODOLOGY

With the Control Program for Engineering Synthesis and Constrained Function Minimization (COPES/CONMIN) optimizing scheme, a nonlinear optimization program is available that is capable of optimizing a wide class of engineering problems [22,23]. Therefore, for the heat exchanger design problem, it was necessary to develop a subroutine, which given a starting design, would analyze an air-cooled heat exchanger, and which would be suitable for coupling with the optimizer, COPES/CONMIN.

Figure 1 illustrates the procedure by which the heat exchanger was analyzed. Initial input consisted of a complete listing of all design parameters, whether known or estimated. Those that were estimates, i.e. unknowns, were later passed to the optimizer as design variables.

The analysis scheme and the optimizer will be discussed in much greater detail in the following chapters.

D. OBJECTIVES

The objectives of this thesis are two-fold.

The first objective was to develop a computer code, hereafter referred to as ANALIZ, which would analyze an air-cooled heat exchanger given any initial design. The analysis scheme was to: be iterative-free, take into account the variation of the heat transfer coefficients and the pressure drop with temperature and/or length of flow path, and finally, be written in such a manner that it could be coupled with an existing optimizer COPES/CONMIN.

The second objective was to actually couple ANALIZ with a numerical optimization program. This would produce a detailed design program which would have the capability to determine an optimum surface, while optimizing the objective function such as size, weight, cost, etc.

II. NUMERICAL OPTIMIZATION

A. BACKGROUND

Almost all design problems require either the maximization or the minimization of some parameter or function. This parameter shall be called the design's objective function [24]. For example, the problem may call for a heat exchanger with a minimum volume. The expression for volume would be the design objective function. For the design to be acceptable, it must satisfy certain design constraints. For example, an air heater must be designed so that it will fit into a given space. Therefore, the engineer must set design constraints on the maximum size of the exchanger.

If the objective function could be easily formulated analytically, the maxima or minima could be found by using the methods of differential calculus. However, the limitations of this method are obvious.

Another numerical method that would be satisfactory for small scale problems would be an iterative solution technique. A computer program could be written containing a series of nested iteration loops that would vary the design parameters and solve the problem for a variety of values for each of the parameters. For other than small, easily formulated problems, the cost in central processor (CPU), time would be prohibitive.

Over the last twenty years, many numerical optimization techniques have been developed specifically for computer

utilization. These techniques usually do not require a specific algebraic equation, but rather any computer algorithm to which design variables can be input and from which the objective function and design constraint values can be determined is acceptable. For this reason, nonlinear programming methods were chosen for the air-cooled heat exchanger design. Some of these techniques were summarized by Shah et al. [25] in figure 2.

1. <u>One-Dimensional Search Methods</u>. Two of the most common of these search methods are the golden section [26] and quadratic interpolation [27]. The former isolates the minimum in regions of successively decreasing size, the latter performs a series of iterations approximating the objective function as a quadratic.

2. <u>Multidimensional Unconstrained Search Methods</u>. These unconstrained searches can be performed by a sequence of one-dimensional minimizations in the proper directions.

3. <u>Multidimensional Constrained Search Methods</u>. A common method here for enforcing the constraints in an optimization scheme are based on the sequential penalty function method. These techniques convert the constrained optimization problem into a sequence of unconstrained problems. This is accomplished by applying either an exterior or interior penalty to the objective function. The Complex Method [15], locates the optimum based on an intuitive approach in a n-dimensional space defined by the independent design variables. The method of feasible directions is used primarily for inequality constraints.

An optimization program based upon the Augmented Lagrangian Multiplier Method and the method of feasible directions was chosen for this research project.

B. CONSTRAINED FUNCTION MINIMIZATION (CONMIN)

Vanderplaats [22], developed an optimization program, CONMIN, based on the method of feasible directions which is

capable of optimizing a wide variety of engineering problems. CONMIN is a FORTRAN program, in subprogram form, that optimizes a function subject to a set of inequality constraints.

The following definitions will be useful in the following discussion:

1. Design Variables - those parameters which the optimization program can change in order to improve the design.

2. Design Constraints - those parameters which must not exceed given bounds for the design to be acceptable.

3. Objective Function - the parameter which is going to be minimized or maximized.

The general nonlinear inequality constrained optimization problem can be written mathematically as follows [28]:

$$\text{Minimize} \quad F(\overline{X}) \qquad (1)$$

Subject to:

$$g_j(\overline{X}) \leq 0 \qquad j = 1,\text{NCON} \qquad (2)$$

$$X_i^{\ell} \leq X_i \leq X_i^{u} \quad k = 1,\text{NDV} \qquad (3)$$

where

$$\overline{X} = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_{NDV} \end{bmatrix} \qquad (4)$$

The vector $\overline{X}$ is the vector of design variables, with NDV equal to the number of design variables. The objective function,

$F(\overline{X})$, given by eq. (1), as well as the constraint functions given by eq. (2), may be linear or nonlinear functions of the design variables. They shall also be explicit or implicit functions of $\overline{X}$, but must have continuous first derivatives. NCON is the number of constraints. X_i^ℓ and X_i^u are the lower and upper bounds or side constraints placed on the design variables. Side constraints could be included in eq. (2), but are treated separately for efficiency. Equality constraints are not dealt with by CONMIN, but will be treated separately by a multiplier method.

CONMIN requires that an initial vector of design variables, $\overline{X}$, which may or may not yield a feasible design, be specified. The design process continues iteratively as:

$$\overline{X}^{q+1} = \overline{X}^q + \alpha^* \overline{S}^q \tag{5}$$

where $\overline{S}^q$ is a vector search direction, α^* is a scalar quantity which determines the amount of change in $\overline{X}$ and q is the iteration number. At iteration q a direction $\overline{S}^q$ must be found which will reduce the objective (usable direction for minimization), without violating any constraints (feasible direction), see Figure 3 [30]. Once $\overline{S}^q$ is determined, eq. (5) becomes a one-dimensional search problem in whcih α^* must be found such that; $F(\overline{X})$ is at a minimum (see Figure 4 [29]), a new constraint is encountered, or a currently active constraint ($g_j(\overline{X}) = 0$) is encountered again.

The design problem at iteration q+1 becomes one of finding a usable-feasible direction, $\overline{S}^q$, and a move parameter

α^*. This process is illustrated geometrically by Johnson [30]. Consider a condenser problem with just two design variables, X_1 and X_2, where

X_1 = condenser tube outside diameter,

X_2 = tube pitch to diameter ratio.

Let the objective function be condenser volume, VOL $(\overline{X})$. Assume that the tube bundle diameter must be greater than a given value, BD_{min}, and that the cooling water pumping power must be less than a given value HP_{max}. Figure 5 illustrates the design problem geometrically.

It should be reiterated here, that while Johnson's example starts with a feasible initial design, (A), this is not a requirement and CONMIN is capable of optimizing given an infeasible initial design. This is obviously a great benefit.

The optimization process begins by calculating the gradient of the objective function by using finite difference. Each design variable is perturbed by .01 in a single forward step.

The gradient of the objective function, $\overline{\nabla}F$, shown in Figure 6 is simply the vector of the first partial derivatives with respect to the design variables; that is:

$$\overline{\nabla}F(\overline{X}) \equiv \overline{\nabla}\,\mathrm{VOL} = \begin{bmatrix} \dfrac{\partial F}{\partial X_1} \\ \\ \dfrac{\partial F}{\partial X_2} \end{bmatrix} \doteq \begin{bmatrix} \dfrac{\Delta \mathrm{VOL}}{\Delta X_1} \\ \\ \dfrac{\Delta \mathrm{VOL}}{\Delta X_2} \end{bmatrix} \qquad (6)$$

Therefore, because no constraints are active or violated at (A). $\overline{\nabla}F$ defines the direction of steepest ascent. Because it is desired to minimize F, the greatest improvement can be made by moving in the negative gradient direction so that

$$\overline{S} = - \overline{\nabla}F = - \overline{\nabla}VOL . \qquad (7)$$

With the value of $\overline{S}$ now determined, a search is performed from (A) until the minimum F is found at (B) on Figure 6. This is accomplished by taking several values of $\overline{X}^{q+1}$ in eq. (5) and interpolating for the α^* which will give the minimum value of F.

The second design iteration is begun at (B) by again perturbing $\overline{X}$ to find $\overline{\nabla}F$. Instead of moving in the direction of steepest descent, a conjugate direction, developed by Fletcher and Reeves [31], is chosen with this method, $\overline{S}$ is calculated as follows:

$$\overline{S}^q = - \overline{\nabla}F(\overline{X})^q + \frac{|\overline{\nabla}F(\overline{X})^q|^2}{|\overline{\nabla}F(\overline{X})^{q-1}|^2} S^{q-1} , \qquad (8)$$

see Figure 7.

The Fletcher-Reeves method is used in order to speed convergence. With the new conjugate direction, a search is performed in this direction until a constraint is encountered. This occurs at (C) of Figure 8 on the pumping power constraint.

At (C), with the HP constraint active, not only is ∇F found, but the gradient of the active constraint is also computed, again using finite difference. The requirements on

the new search direction are now twofold; it must reduce the objective function and, at the same time, not violate the active constraint. This is solved by using the method of feasible directions developed by Zoutendijk [32], and implemented by Vanderplaats and Moses [33].

The problem of finding the new $\bar{S}$ can be stated as [34]:

Maximize β subject to the constraints

$$\bar{\nabla}F(\bar{X}) \cdot \bar{S} + \beta \leq 0 \tag{9}$$

$$\bar{\nabla}g_j(\bar{X}) \cdot \bar{S} + \theta_j\beta \leq 0 \qquad j = 1,NAC \tag{10}$$

$$\bar{S} \cdot \bar{S} \leq 1 \tag{11}$$

where $\bar{\nabla}g_j(\bar{X}) = -\bar{\nabla}HP$ and NAC is the number of active constraints (in this case NAC = 1).

If equation (9) is satisfied and β is positive, the search direction will reduce the objective function and is defined as a usable. If equation (10) is satisfied and β is positive, $\bar{S}$ is a feasible direction, because in this direction, no constraints will be violated if only a small move is taken. See Figure 3. θ_j is defined as the push-off factor for the active constraint and causes the design to move away from the constraint. θ_j must be greater than or equal to zero in order to maintain a feasible design. If the maximum value of β from equations (9) through (11) is zero, then there is no direction that will both reduce the objective function and also be feasible. Therefore, the current design is, at least, a local

minimum. In Johnson's example, a usable-feasible direction exists and a one-dimensional search leads to D in Figure 8 where the minimum bundle diameter, BD_{min}, constraint is met.

From D it should be noted here that CONMIN had information regarding the linearity of the BD_{min} constraint and, therefore, in (10), has set $\theta_j = 0$ to allow $\overline{S}$ to follow the constraint as shown in Figure 9. The one-dimensional search along this constraint is carried out until no further design improvement is realized. This occurs at E .

This discussion of CONMIN would not be complete without citing the program's limitations. NDV directly affects the computational time required to reach the optimum. Since the calculation of gradients required for each design variable at the beginning of each design iteration is found by a finite difference step, which requires a complete pass through the analysis portion of the program, there is a subsequent increase in CPU time as NDV increases. Also, due to the interaction between design variables, as NDV gets larger, convergence slows during the optimization process. Vanderplaats [24], recommends that for most problems of general interest a practical limit of NDV = 20 be imposed. NCON does not present the same problem because gradient information is calculated simultaneously with ∇F and then only if the constraint is active or violated.

CONMIN offers no guarantees that a global minimum has been reached. Therefore, to lend some assurance, the design is

started with several different initial vectors until the same optimal design is reached.

Although CONMIN performs very well with inequality constraints, equality constraints such as:

$$h_K(\overline{X}) = 0$$

cannot be dealt with directly, but must be treated separately, using a different method, which will be discussed in the following sections.

C. CONTROL PROGRAM FOR ENGINEERING SYNTHESIS (COPES)

Recall that CONMIN was written in subroutine form, Vanderplaats [23], has developed a main program which greatly enhances the use of CONMIN.

For this main program, COPES, the user must supply an analysis subroutine titled ANALIZ. The subroutines CONMIN and ANALIZ are then used by COPES to optimize the objective function subject only to the inequality constraints.

ANALIZ must be organized into three segments: input, analysis and output. Based on the value of a counter, ICALC, ANALIZ performs the proper function in sequence.

The COPES program currently provides four specific capabilities:

1. Single Analysis - one cycle through the program, as if ANALIZ was executing alone.
2. Optimization - minimization or maximization of the objective function with constraints and side-constraints imposed.

3. Sensitivity Analysis - used to explore the effect of changing one or more design variables on one or more functions.

4. Two Variable Function Space - provides tables of data of all specified combination of two design variables.

However, a recent addition to COPES (still in the developmental stages), has put the Augmented Lagrangian Multiplier Method (ALMM) at the disposal of the programmer. Because of its good rate of convergence and its theoretical properties, the ALMM is preferred for equality constrained problems [35].

Therefore, COPES will call the various subroutines in order to optimize the objective function; subroutine ANALIZ for necessary analysis information, CONMIN for an optimum based on inequality constraints, and a subroutine (yet to be named), utilizing ALMM for an optimum satisfying the equality constraints.

For detailed explanation of the ALMM, its background and mathematical derivation, consult reference [35]; but for now consider the equality constrained problem:

$$\text{Min} \quad f(\bar{X}) \tag{10a}$$

$$\text{Subject to} \quad h_k(\bar{X}) = 0 \quad k = 1,\text{NECON} \tag{10b}$$

where NECON is the number of equality constraints.

Define the "modified" Lagrangian function as:

$$L(\bar{X},\bar{\lambda}) = f(\bar{X}) + \sum_{i=1}^{m} \lambda_i \, h_k(\bar{X})$$

where λ_i is the Lagrangian multiplier. The problem can now be stated as:

$$\text{Min} \quad L(\bar{X},\bar{\lambda}) \tag{10c}$$

$$\text{Subject to} \quad h_k(\bar{X}) = 0 \tag{10d}$$

Then, according to Lagrange, if a $\bar{\lambda}$ can be found for which $\bar{X}$ solves the problem stated above, then $\bar{X}$ is also the solution to the original problem, eq. (10a) and (10b).

The new problem is solved by the conventional exterior penalty function method, because this is believed to be one of the most efficient algorithms for the solution of such equality constrained problems [35].

II. HEAT EXCHANGER ANALYSIS

A. INTRODUCTION

To meet the objectives of this thesis, an analysis program for an air-cooled heat exchanger must be coupled with a numerical optimization scheme to produce a complete, detailed design package. COPES/CONMIN has greatly simplified this task.

This analysis program must be written in subroutine form, titled ANALIZ, and organized into three segments: input, execution, and output. The analysis subroutine must also:

1. Take into account the variation of the heat-transfer coefficients and differential pressure drop with temperature and/or length of flow path.
2. Be iterative free, if possible.
3. Be written in such a manner that the optimizer will play a role in surface selection.

With the number of design variables approaching the practical limit, the importance of an iterative free analysis subroutine cannot be over-emphasized. The reason being, that at the beginning of each design iteration in CONMIN, the calculation of all gradients (each design variable and active constraint) requires a complete pass through ANALIZ. Therefore, the computational time required by ANALIZ directly affects the time required to reach the optimum.

B. PROBLEM FORMULATION

The air-cooled heat exchanger is shown in Figure 10. A cross-flow arrangement with both fluids unmixed was chosen.

Cool air enters the heat exchanger at temperature T_{c_1}, pressure p_∞, and constant specific heat c_{p_a}. The cool air makes one pass through the exchanger as it flows over an isosceles pitched bank of finned tubes. The air is heated by water, in single phase, at an entering temperature of T_{h_1}, and constant specific heat c_{p_w}.

The analysis of the air-cooled, cross-flow heat exchanger centers about the first law of thermodynamics and on the heat transfer equation. These equations as they apply to the exchanger of Figure 10 are as follows:

$$\dot{Q}_3 = \dot{m}_a c_{p_a} (T_{c_2} - T_{c_1}) \qquad (12)$$

$$\dot{Q}_4 = \dot{m}_w c_{p_w} (T_{h_1} - T_{h_2}) \qquad (13)$$

$$\dot{Q}_5 = U_m \, A \, \Delta T_m \qquad (14)$$

where $\dot{m}_a$, T_{c_2} and $\dot{m}_w$, T_{h_2} are the fluid mass flow rates and exit temperatures of air and water respectively. U_m is the true mean overall heat transfer coefficient based on the outside root tube area, A is the total heat transfer surface area of the exchanger used to compute U_m, and ΔT_m is the mean temperature difference of the given exchanger.

The object of the analysis, therefore, is to determine $\dot{Q}_3$, $\dot{Q}_4$ and $\dot{Q}_5$ given an initial listing of values for the design parameters. The list includes the following:

Tubeside mass flow rate, lbm/hr

Entrance temperature of hot stream, °F

Exit temperature of hot stream, °F

Specific heat of hot fluid, BTU/lbm - °F

Air mass flow rate, lbm/hr

Entrance air temperature, °F

Exit air temperature, °F

Entrance air pressure, psi

Specific heat of air, BTU/lbm-°F

Tube inside diameter (ID), in.

Tube outside diameter (OD), in.

Fin height, in.

Fin thickness, in.

Fin spacing, in.

Transverse Pitch, in.

Longitudinal Pitch, in.

Bank height, in.

Bank width, in.

Cross-flow arrangement

Fin type

Number of rows

Number of passes

Given heat transfer rate, BTU/hr

Thermal Conductivity of tube material, BTU/hr.-ft.-°F

Thermal Conductivity of fin material, BTU/hr.-ft.-°F

Among the initial listing of design parameters above, there are parameters that are known and will remain constant

throughout the design problem. Also, there will be those parameters that are unknown and can vary, i.e. design variables. On the way to determining the various heat transfer rates, other information will have been computed. This information includes the objective function, constraining functions, and other design data, such as the number of tubes per vertical row.

The optimizer will then manipulate the design variables in order to find an optimum, while at the same time, performing a heat balance, that is:

$$\dot{Q} = \dot{Q}_3 = \dot{Q}_4 = \dot{Q}_5 \qquad (15)$$

where $\dot{Q}$ may be some given heat transfer rate.

C. PERFORMANCE CALCULATION PROCEDURE

With the temperatures, mass flow rates, and specific heats all specified in the listing of design parameters; whether they be constant or variable, the only unknown quantities on the right hand side of equations(12) through (14), are U_m, A, and ΔT_m. They will be determined as shown in Figure 1.

1. Mean Temperature Difference (MTD)

For many flow arrangements, various approahces for determining MTD, mainly using diagrams, are available [36], which have proven very useful in manual design efforts. For computerized design, however, an explicit, approximate equation is desirable in order to achieve a fast, sufficiently accurate

calculation of the mean temperature difference of a given flow arrangement.

Roetzel, et al. [37], presented such an approximate equation together with empirical coefficients for nine counter-current cross-flow arrangements as they apply to air-cooled heat exchangers.

Roetzel used the familiar equation for the MTD of the given flow arrangement ΔT_m:

$$\Delta T_m = F \cdot \Delta T_{\ell_m} \tag{16}$$

where ΔT_{ℓ_m} is the limiting case of pure countercurrent flow:

$$\Delta T_{\ell_m} = \frac{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})}{\ln \dfrac{(T_{h_1} - T_{c_2})}{(T_{h_2} - T_{c_1})}} \tag{17}$$

and F is a correction factor determined by a different set of coefficients for each flow arrangement. Roetzel reported the following function suitable for F:

$$F = 1 - \sum_{i=1}^{m} \sum_{k=1}^{n} a_{i,k} (1 - \nu_{1_m})^k \sin(2\, i \arctan R) \tag{18}$$

where ν_{1_m} is the dimensionless LMTD.

$$\nu_{1_m} = \frac{\Delta T_{\ell_m}}{T_{h_1} - T_{c_1}} \tag{19}$$

$$R = \frac{T_{h_1} - T_{h_2}}{T_{c_2} - T_{c_1}} \qquad (20)$$

and the coefficients $a_{i,k}$ of the approximating equation (18), were calculated using a standard least squares estimation program [38], and are reported in reference [37]. The assumption that both streams were unmixed was used in their calculation.

When more than four tubeside passes are used, it is assumed that the heat exchanger has approached the limiting case of pure counterflow and F is set equal to one [2].

Having determined the MTD, the remainder of the analysis procedure follows Roetzel's [39] general approximation method for determining the mean overall heat transfer coefficient, U_m, for any flow arrangement while taking into account the variation of the heat transfer coefficients and the pressure drop with temperature and/or length of path.

Before continuing with specific analysis procedures, a brief summary of Roetzel's general approximation method is in order.

The local overall heat transfer coefficient based on the outside root tube area can be written as follows:

$$U = \frac{1}{\frac{A_o}{A_i}\frac{1}{H_i} + \frac{A_o \ln(r_o/r_i)}{2\pi kL} + \frac{1}{H_o \eta_f}} \qquad (21)$$

where η_f is the efficiency of the extended surface.

In order to determine the individual convection heat transfer (film) coefficients, H_i and H_o, according to the

conventional methods, the coefficients would be considered constant, and the necessary fluid properties for their calculation would be evaluated at some mean bulk temperatures, T_{h_b} and T_{c_b}.

However, the film coefficients are not constant, but vary with temperature and/or length of flow path. Roetzel has taken these variations into account with the use of corrected reference temperatures. Two sets of corrected reference temperatures are determined: T_{h_I}, T_{c_I} and $T_{h_{II}}$, $T_{c_{II}}$. Therefore, for each set of corrected reference temperatures, the film coefficients are determined in the conventional manner using the reference temperatures in place of the bulk temperatures.

With the film coefficients, H_{i_I}, H_{o_I}, $H_{i_{II}}$ and $H_{o_{II}}$, two local overall heat transfer coefficients can be calculated from equation (21), U_I and U_{II}.

Finally, the true mean overall heat transfer coefficient is calculated as:

$$\frac{1}{U_m} = \frac{1}{2}\left[\frac{1}{U_I} + \frac{1}{U_{II}}\right] \tag{22}$$

2. Determination of Reference Temperatures

The reference temperatures for a pure counterflow heat exchanger must first be determined from [39]:

$$T'_{h_I} = T_{h_2} + (T_{h_1} - T_{h_2}) \left[\frac{\Delta T_I - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})}\right] \tag{23}$$

$$T'_{h_{II}} = T_{h_2} + (T_{h_1} - T_{h_2}) \left[\frac{\Delta T_{II} - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (24)$$

$$T'_{c_I} = T_{c_1} + (T_{c_2} - T_{c_1}) \left[\frac{\Delta T_I - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (25)$$

$$T'_{c_{II}} = T_{c_1} + (T_{c_2} - T_{c_1}) \left[\frac{\Delta T_{II} - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (26)$$

where

$$\Delta T_I = (T_{h_1} - T_{c_2})^{.78868} \cdot (T_{h_2} - T_{c_1})^{.21132} \quad (26a)$$

$$\Delta T_{II} = (T_{h_1} - T_{c_2})^{.21132} \cdot (T_{h_2} - T_{c_1})^{.78868} \quad (26b)$$

Equations (26a) and (26b) were derived by Roetzel in references [40] and [41]. For the special case where the fluid heat capacity rates, $\dot{C}$, are equal, that is,

$$\dot{m}_h \, c_{p_h} = \dot{m}_c \, c_{p_c}$$

the term

$$\frac{\Delta T_j - (T_{h_2} - T_{c_1})}{(T_{h_2} - T_{c_2}) - (T_{h_2} - T_{c_1})}$$

of equations (23) through (26), with j = I and II, becomes .78868 or .21132 respectively.

3. Correction of Reference Temperatures for Given Arrangement

With inlet and outlet temperatures fixed, pure counterflow yields the highest mean temperature difference. Therefore, for any other arrangement, the temperature difference would be smaller. Thus, the corrections are applied in the following manner [39]:

$$T_{h_I} = T'_{h_I} - \psi_{h_I}$$

$$T_{h_{II}} = T'_{h_{II}} - \psi_{h_{II}}$$

$$T_{c_I} = T'_{c_I} + \psi_{c_I}$$

$$T_{c_{II}} = T'_{c_{II}} + \psi_{c_{II}}$$

where ψ_{h_k} and ψ_{c_k} are the temperature corrections of the hot and cold streams, respectively. The corrections are calculated as follows:

$$\psi_{h_j} = \Delta T_j \left[\frac{1 - \Delta T_m / \Delta T_{\ell_m}}{1 + (\dot{C}_h / \dot{C}_c)^{2/3}} \right]$$

$$\psi_{c_j} = \Delta T_j \left[\frac{1 - \Delta T_m / \Delta T_{\ell_m}}{1 + (\dot{C}_c / \dot{C}_h)^{2/3}} \right]$$

The corrected reference temperatures are now used to determine the thermal conductivities and absolute viscosities of

fluids for later use in the calculation of the film coefficients. Thermal conductivity and viscosity data are usually presented in tabular form. However, for use on the computer, an explicit, simple approximate equation with temperature as the independent variable, was desirable. Water and air were chosen as two fluids that were likely to be involved in an air-cooled heat exchanger design. Figures 11, 12 and 13 indicate that the thermal conductivities of air and water and the viscosity of air can be approximated by a second order polynomial. The viscosity of the hot tubeside fluid, water, must be treated specially, due to the following considerations.

Calculations for the tubeside heat transfer coefficient in the laminar, transition and turbulent regions all involve the correction, $(\mu/\mu_{wall})^{.14}$ [42]. In the past, an iterative procedure was required to determine the inside tube wall temperature in order to evaluate μ_{wall}. Roetzel [43], has developed an iteration-free method for determining this correction. Roetzel's method assumes that the tubeside fluid's viscosity follows Andrade's Correlation, that is:

$$\mu = \alpha e^{\beta/T}$$

Using the viscosity data of saturated water from reference [44], the coefficients were determined through regression analysis with the resulting equation:

$$\mu_w = .01339 \exp(\frac{2715.7764}{T}) \qquad (27)$$

where temperature is in degrees Rankine and viscosity is in lbm/ft-hr. Equation (27) was plotted next to the experimental data with results that indicate that water does indeed satisfy Andrade's equation, see Figure 14.

4. Uncorrected Tubeside Heat Transfer Coefficient

The tubeside heat transfer coefficient is calculated from one of three Nusselt-type empirical equations [42], as follows:

For laminar flows, Reynolds Number < 2,100

$$\frac{HD_i}{k} = 1.86\ Re^{1/3}\ Pr^{1/3}\ \left(\frac{D_i}{L}\right)^{1/3}\ \left(\frac{\mu}{\mu_{wall}}\right)^{.14} \tag{28}$$

For transition regions, 2,100 < Re < 10,000

$$\frac{HD_i}{k} = [Re^{2/3} - 125][1 + \left(\frac{D_i}{L}\right)^{2/3}]\ Pr^{1/3}\ \left(\frac{\mu}{\mu_{wall}}\right)^{.14} \tag{29}$$

For turbulent flow, Re > 10,000

$$\frac{HD_i}{k} = .023\ Re^{.8}\ Pr^{1/3}\ \left(\frac{\mu}{\mu_w}\right)^{.14} \tag{30}$$

Therefore, before any calculations can even begin, the Reynolds Number, Re, must be computed to determine the type of flow.

The Reynolds Number will be calculated as:

$$Re_j = \frac{D_i\ \dot{m}_h}{A_x\ \mu_{h_j}}$$

D_i, $\dot{m}_h$ is supplied in the initial list of parameters and μ_h is calculated from equation (27). This leaves only the cross-sectional flow area, A_x, to be determined, where:

$$A_x = N_p \frac{\pi D_i^2}{4}$$

The number of tubes per pass, N_p, is determined geometrically, having been given the tube bank height, number of rows, number of passes, transverse pitch, diameters, and fin height initially. Figure 15 details the procedure of finding N_p. Mathematically,

$$N_p = [\frac{n_{rows}}{n_{pass}}][h - D_f - \frac{P_t}{2}]/P_t$$

and would be a rational number. This rational number is then truncated to an integer number of tubes.

With Re calculated, the Prandtl Number,

$$Pr_j = \frac{C_p \mu_{h_j}}{k_{h_j}}$$

is computed from given and previously determined thermo-physical properties.

The uncorrected tubeside heat transfer coefficients, H'_{i_j}, that is; H_{i_j}, without the factor $(\mu/\mu_w)^{.14}$, can be calculated from the proper choice of equations (28) through (29).

5. Calculation of Wall and Associated Resistances

Equation (21) can be written in a more general form as:

$$U = \frac{1}{R_i + R_{wall} + R_o} \tag{31}$$

where R_i, R_w and R_o are the inside wall and outside heat transfer resistances respectively. Additional resistances, such as contact or fouling, can also be added here.

In comparing equations (21) and (31), the resistances can be computed as:

$$R_{i_j} = \frac{D_o}{D_i} \frac{1}{H_{i_j}} \tag{31a}$$

$$R_{wall} = \frac{r_o \ln(D_o/D_i)}{k_{wall}} \tag{31b}$$

$$R_{o_j} = \frac{1}{H_{o_j} \eta_f} \tag{31c}$$

6. Calculation of Airside Heat Transfer Coefficients

In order to have the optimization program play a significant role in the selection of an optimized surface for a finned tube heat exchanger (which is one of the objectives of this project), an explicit equation for H_o involving tube and bank design parameters as independent variables is a necessity.

In the past, comparison methods, as described by Shah [45], were used to choose the best surface from a list of

surfaces for which experimental heat transfer and friction data existed. The data is presented in graphical form, where Friction and Colburn Factors are plotted as a function of Reynolds Number. With J, $\dot{m}_a$, c_p and Pr known, the film coefficient can be computed.

Therefore, in previous heat exchanger optimization programs, a given surface described by its pitch, outside fin diameter, fin thickness, fin spacing, and outside tube diameter, had to be chosen beforehand. After the surface configuration had been selected, expressions for f and J were obtained by fitting polynomials to the experimental data described earlier.

Briggs and Young [47] provide a means of getting past this obstacle with an improved convection heat transfer correlation for air flowing across triangular pitch banks of high finned tubes.

Briggs and Young expanded the work of Ward and Young [48], to cover a total of 18 differently configured finned tube banks in order to determine the effect of fin thickness and tube pitch on the airside heat transfer coefficient. The heat transfer data for the high-fin tube banks were correlated to give:

$$Nu = \frac{H_{o_j} D_o}{k_j} = .1378 \, Re^{.718} \, Pr^{1/3} \left(\frac{s}{\ell}\right)^{.296} \qquad (32)$$

where s is the distance between adjacent fins and ℓ is the fin height. Equation (32) is based on tubes having a wide range

of fin heights, fin thicknesses, fin spacing and outside tube diameter and can be used to predict H_{o_j} for a bank of tubes six rows deep. Figure 11 of reference [48] is used to correct H'_{o_j} for banks of other than six rows.

7. Calculation of Fin and Surface Efficiencies

In order to calculate the outside heat transfer resistance, which will be used to calculate the correction for the tubeside heat transfer coefficient and finally, the local overall heat transfer coefficients, the extended surface efficiency, η_f, must be computed.

The surface efficiency accounts for the temperature drop from the root to the tip of the fin, due to the thermal resistance of the fin material. Thus, even though the heat transfer has been increased by the additional area of the extended surface, the area is not as effective as if it were to be all at the root temperature.

The surface efficiency can be expressed as [49]:

$$\eta_f = 1 - \frac{A_f}{A}(1-\phi)$$

where ϕ, the fin efficiency for a radial fin, can be found from [49]:

$$\phi = \frac{2\, r_o}{m(r_f^{\,2} - r_o^{\,2})}\left[\frac{I_1(mr_f)k_1(mr_o) - k_1(mr_f)I_1(mr_o)}{I_o(mr_o)k_1(mr_f) + I_1(mr_f)k_o(mr_o)}\right]$$

The finned area, A_f, and the total heat transfer area, A, are computed as follows:

$$A_s = N_T\, w\, \eta_f\, \frac{\pi}{2}(D_f^{\,2} - D_o^{\,2})$$

$$A = N_T w\left[\frac{\eta_f \pi}{2}(D_f^{\,2} - D_o^{\,2}) + (1 - \eta_f t) D_o \pi\right]$$

With R_o calculated from equation (31c), the correction to the tubeside film coefficient can be made.

8. <u>Correction of the Tubeside Heat Transfer Coefficient</u>

With the tubeside heat transfer coefficient dependent on the wall temperature, the dependence has either been neglected, or the wall temperature has been calculated with an iterative technique in the past. Roetzel [43], has proposed an improved iterative-free method for finding the "Seider-Tate" correction, $(\mu/\mu_{wall})^{.14}$, when the tubeside fluid obeys Andrade's viscosity correlation.

From Roetzel's formulation:

$$\left(\frac{\mu}{\mu_{wall}}\right)_j^{.14} = -\frac{u_j}{2} + \left(\frac{u_j^{\,2}}{4} + v_j\right)^{1/2}$$

where

$$u_j = .007 \ln\left(\frac{\mu^*_{wall_j}}{\mu_j}\right)\frac{1}{B_j}\left[\frac{1 - T_{c_j}/T_{h_j}}{T_{h_j}/T^*_{wall} - 1}\right] + \frac{T_{c_j}}{B_j T_{h_j}} - 1$$

$$v_j = -.07 \ln\left(\frac{\mu^*_{wall_j}}{\mu_j}\right)\frac{1}{B_j}\left[\frac{1 - T_{c_j}/T_{h_j}}{T_{h_j}/T^*_{wall} - 1}\right] + \frac{T_{c_j}}{B_j T_{h_j}}$$

where

$$B_j = H'_{i_j} A (R_{wall} + R_{o_j})$$

(Note that all temperatures are in degrees absolute.)

All the parameters on the right-hand side of equation (21) are now available. Therefore, the two reference overall heat transfer coefficients, U_I and U_{II}, can be calculated. The mean overall heat transfer coefficient follows easily from equation (22).

With U_m, $\dot{Q}_5$ can be calculated, with the heat balance to be performed by the optimizer.

9. <u>Calculation of Air and Tubeside Pressure Drops</u>

The final calculations before computing objective and constraint functions for the optimization problem involve the pressure drops in the heat exchanger.

The basic equations that will be used for the calculation of the pressure losses are as follows:

a. Tubeside [42]

$$\Delta p_i = \frac{f_i \dot{m}_h^2 w n_p}{2 g_c A_{x_T}^2 \rho D_i (\frac{\mu}{\mu_w})} z + \frac{(n_p - 1) \dot{m}_h^2}{2 g_c A_{x_T}^2 \rho} \qquad (33)$$

where z = .14 below Re = 2100 and z = .25 for Re greater than 2100.

b. Airside [50]

$$\Delta p_o = \frac{f_o n_r \dot{m}_c^2}{A_{ff}^2 g_c \rho} \qquad (34)$$

where A_{ff} is the minimum flow area.

The friction factor for the tubeside flow, f_i, is taken from Figure 9.5 of reference [42], assuming fully developed flow. For use on the computer, an explicit expression for f_i was obtained by fitting a line and an exponential to the experimental data of Figure 9.5. This follows:

$$\underline{Re \leq 1000} \qquad f_i = .5/Re$$

$$\underline{Re > 1000} \qquad f_i = .003215\ Re^{-.2694}$$

Just as in the case of the airside film coefficient, for surface optimization on the computer, an explicit equation for the airside friction factor, f_o, is a necessity. Robinson and Briggs [50], presented such an expression for f_o for air flowing across triangular pitch banks of finned tubes. Robinson and Briggs' work closely parallels that of Briggs and Young [47]. The Robinson-Briggs Correlation:

$$f_o = 18.93\ Re^{-.316} \left(\frac{P_t}{D_o}\right)^{-.927} \left(\frac{P_t}{P_L}\right)^{.515} \tag{34a}$$

covers the range of tube sizes and pitches used in air-cooled heat exchangers [50].

Therefore, with Re_I and Re_{II}, the four reference pressure drops, Δp_{i_j}, and Δp_{o_j}, may be computed from equations (33) and (34). Following Roetzel's general approximation method:

$$\Delta p_i = \left[\frac{\Delta p_{i_I}}{U_I} + \frac{\Delta p_{i_{II}}}{U_{II}}\right] / \left[\frac{1}{U_I} + \frac{1}{U_{II}}\right]$$

For a gas, an additional correction is needed because the density in equation (33) is strongly dependent on pressure, which is changing through the exchanger. Using the inlet pressure as reference:

$$\Delta p_o' = \left[\frac{\Delta p_{o_I}}{U_1} + \frac{\Delta p_{o_{II}}}{U_{II}}\right] / \left[\frac{1}{U_1} + \frac{1}{U_{II}}\right]$$

$$\Delta p_o = p_1 \left[1 - \left(1 - \frac{2\Delta p_o'}{p_1}\right)^{1/2}\right] \qquad (35)$$

All the necessary information from an analysis viewpoint has now been calculated. Functions needed for the numerical optimization process shall follow.

10. Objective and Constraint Functions

The objective functions available for minimization are defined as follows:

a. Volume $= wh[D_f + (n_r - 1) P_L \cos \theta]$

where

$$\theta = \arcsin (P_t/2P_L) \qquad (36)$$

b. Heat Transfer Area $= N_T w\left[\frac{(D_f^2 - D_o^2)\pi}{2\,S} + \left(1 - \frac{t}{S}\right) D_o \pi\right]$

c. Air Horsepower $= \dfrac{\Delta p_a \cdot \dot{m}_a}{\rho}$

d. Airside Pressure Drop

e. Tubeside Pressure Drop

Limitations were imposed on the following inequality constraints in order to keep the design within practical physical bounds:

a. The diameter ratio,

$$\text{DRATIO} = \frac{D_f}{D_o}$$

must be kept reasonable. This can also be accomplished to some extent by placing side constraints on the design variables, ℓ and D_o. See Figure 16.

b. The optimizer must be prevented from driving the tube thickness,

$$\text{TUBTH} = (D_o - D_i)/2$$

to zero.

c. The tubes must be kept from touching in both the longitudinal and transverse directions,

$$\text{TOUCHN} = D_f - P_t$$

$$\text{TOUCHL} = D_f - P_L$$

TOUCHL and TOUCHN, therefore, must be kept below zero.

d. Reasonable temperature profiles must be maintained at both ends of the exchanger. See Figure 10-7 of Ref. [44],

$$\text{PROFH} = T_{c_2} - T_{h_1}$$

$$\text{PROFC} = T_{c_1} - T_{h_2}$$

that is, PROFH and PROFC, must be negative.

e. The number of tubes per vertical row, VROWR, shall not be allowed to go below 2.

f. The free face area, that is, the minimum flow area for air, must obviously be greater than zero,

$$\text{DELSFF} = [\text{projected tube area}] - hw$$

where the projected tube area, STOTAL, is

$$\text{STOTAL} = N_r [D_o w + \frac{D_f\, t\, w}{s}]$$

and N_r is the number of tubes per vertical row. DELSFF must be less than zero.

g. The airside and tubeside pressure drops must be kept within design constraints.

h. From experience, the argument of the square root of equation (35),

$$\text{ARG5} = 1 - \frac{2\Delta p'}{P_1}$$

has been driven below zero. It must therefore be constrained.

i. To maintain an isoscele pitch bank, the angle, Θ, as shown in Figure 17 and as defined in eq. (36), must be constrained. One such constraining value is:

$$\Theta_m = \arccos\ (D_f/2P_L) \qquad (37)$$

The arguments of the arcsin and arcos of eq. (36) and (37) must be constrained from going beyond 1.

j. The heat balance described by eq. (15) is performed by three equality constraints:

$$\text{QRATIO} = \dot{Q}_5/\dot{Q} \qquad (38)$$

$$\text{QRATIO1} = \dot{Q}_3/\dot{Q}_5 \qquad (39)$$

$$QRATIO2 = \dot{Q}_4/\dot{Q}_3 \tag{40}$$

All constraints are set equal to 1.

Besides the constraints described in paragraphs a. through j., above, lower and upper bounds are placed on the design variables to assure a reasonable design.

IV. RESULTS

A. BACKGROUND

Case studies were chosen as the best way to test the capabilities of the program for Heat Exchanger Design using Numerical Optimization (HEDSUP). The design problems posed were made as realistic as possible.

1. Capabilities

HEDSUP currently has the capability to design for nine different configurations of triangular pitch banks of finned tubes:

TYPE 1 - 1 ROW, 1 PASS

TYPE 2 - 2 ROW, 1 PASS

TYPE 3 - 3 ROW, 1 PASS

TYPE 4 - 4 ROW, 1 PASS

TYPE 5 - 2 ROW, 2 PASS

TYPE 6 - 3 ROW, 3 PASS

TYPE 7 - 4 ROW, 2 PASS

TYPE 8 - 4 ROW, 2 PASS

TYPE 10 - PURE COUNTERFLOW

TYPE 10 will include exchangers with a configuration of n rows, n passes, where n can go from five to 20.

The banks must be constructed of high-finned tubes (ℓ > .0625 in. [42]) with the fins having a rectangular profile of constant thickness. Additional profiles can be inserted into HEDSUP quite simply, provided that its fin efficiency can

be expressed explicitly as:

$$\eta = f(\ell, t, H, k)$$

See Subroutine FINEFF of the program listing, Appendix D.

At present, HEDSUP can provide the design parameters for an air-cooled heat exchanger optimized for any one of the following design objectives:

(a.) Minimum Volume

(b.) Minimum Heat Transfer Surface Area

(c.) Minimum Air Horsepower

(d.) Minimum Airside Pressure Drop

(e.) Minimum Tubeside Pressure Drop

Additional design objectives can be used, provided that they can be expressed explicitly as a function of the design variables and they are added to the common block. *It should also* be pointed out that any design variable may simultaneously be a design objective as long as it conforms to the restrictions of both. For example, an exchanger may be designed for minimum bank height.

The airside fluid is restricted to dry air. The tubeside fluid is presently limited to water in single phase. Other tubeside fluids can be used by HEDSUP, provided that their visocisites obey Andrade's Law and the fluid's thermal conductivities can be expressed explicitly as a function of temperature. The fluid's specific gravity would also have to be placed in the denominator of eq. (33).

B. CASE STUDIES

1. Case One

a. Problem Formulation

An air-cooled heat exchanger is to be designed for minimum volume with a heat transfer rate of 10,000,000 BTU's per hour. Water is to be cooled from 200°F to 125°F. Dry air will enter the exchanger at 95°F and leave at 130°F. Specifications call for a fan that can produce a pressure difference of two inches of water.

b. Design Variable Framework

From the list of design parameters in Section III.B, the design variables must be singled out, identified to COPES, and given side constraints. All parameters must be given an initial value. Only the values of the design variables will change.

Assuming constant specific heats,

$$c_{p_w} = 1.0 \quad \text{BTU/lbm-°F}$$

$$c_{p_a} = .24 \quad \text{BTU/lbm-°F}$$

the mass flow rates of both fluids can be determined from eqs. (12) and (13), because the heat transfer rate and temperature differences are given,

$$\dot{m}_w = \frac{\dot{Q}}{c_{p_h} \Delta T_n} = 133333. \quad \text{lbm/hr}$$

$$\dot{m}_a = \frac{\dot{Q}}{c_{p_c} \Delta T_c} = 190476.2 \quad \text{lbm/hr}$$

A cross-flow arrangement, fin profile, tube material, and fin material must be chosen.

The design variables for this example are, therefore:

$$.232 < D_i < 2.325 \text{ in.} \qquad D_i^{\,i} = 2.0 \text{ inches}$$

$$.24 < D_o < 2.5 \text{ in.} \qquad D_o^{\,i} = 2.5 \text{ inches}$$

$$.0625 \text{ in.} < \ell < \infty \qquad \ell^{i} = .46 \text{ inches}$$

$$.01 < t < .0235 \text{ in.} \qquad t^{i} = .023 \text{ inches}$$

$$.08 < S < .125 \text{ in.} \qquad S^{i} = .111 \text{ inches}$$

$$0.0 < P_L < 4.0 \text{ in.} \qquad P_L^{\,i} = 2.125 \text{ inches}$$

$$0.0 < P_t < 4.0 \text{ in.} \qquad P_t^{\,i} = 4.0 \text{ inches}$$

$$0.0 < w < 500 \text{ in.} \qquad w^{i} = 490 \text{ inches}$$

$$0.0 < h < 500 \text{ in.} \qquad h^{i} = 350 \text{ inches}$$

The side constraints on the design variables are of a practical nature with the exception of the lower bounds on fin height. Recall that the use of eq. (32) is restricted to high-fins. High-fins will also tend to keep the fluid unmixed, which was an assumption used when defining the coefficients $a_{i,k}$, used in eq. (18).

c. Constraint Framework

From the problem statement, the airside pressure drop must be less than two inches of water or .0722 psi,

$$0 < \Delta p_a < .0722 \text{ psi}$$

From a practical standpoint:

$$0.0 < \Theta < 1.3$$

$$1.0 < \text{DRATIO} < 2.5$$

$$.018 < \text{TUBTH} < .18 \text{ inches}$$

$$-\infty < p_w < .14 \text{ psi}$$

The equality constraint

$$\frac{\dot{Q}_5}{\dot{Q}} = = 1.0$$

where

$$\dot{Q}_5 = U\ A\ \Delta T_m$$

$$\dot{Q} = 10{,}000{,}000 \text{ Btu/hr}$$

will satisfy the heat balance.

The bounds on TOUCHN, TOUCHL ARG5, ARG7, ARG8, DELSFF and VROWR were discussed in Section III.C.10. Constraints on PROFH and PROFC were unnecessary because all temperatures were specified in the problem statement.

As with design variables, constraints must be identified to COPES by location in the common block. See the global catalog, Appendix A.

d. Methodology

Ideally, a three dimensional design matrix can now be constructed of optimum exchanger designs with minimum volumes. The matrix would be constructed by first holding the tube and fin materials constant and varying the configuration, i.e. Type 1, Type 2, Type 3, etc. Next the tube material would be varied with the fin material and exchanger configuration held constant and so forth. However, for case study one, the tube material will be chosen as copper, k = 200 BTU/ft-hr-°F, and the fin material will be aluminum, k = 118 BTU/ft-hr-°F.

Also, in order to simulate an actual trade off study, the constraint framework will be fixed throughout the individual case studies.

Problems arise in constructing the matrix when trying to determine the true minimum volume design for each configuration. Unfortunately, the choice of initial design parameters (starting point), coupled with the input parameters for ALMM, will sometimes lead to entirely different optimum designs with volumes differing by over 100%.

The ALMM parameters include the initial multiplier, CC, the multiplication factor, CMULT, and the maximum multiplier value, CCMAX. Experience has shown that setting

$$\text{CMULT} = 2.0$$
$$\text{CCMAX} = 1000.$$

will suffice for almost all problems. However, there is much "artwork" involved with the choice of CC. From experience,

an initial multiplier of 10 works well when starting far from the equality constraint, i.e.:

$$.8 < \text{QRATIO} < 1.2$$

However, when approaching the heat balance, i.e., the equality constraint, a CC = 100 is necessary for the heat balance to converge. Equation (15) is considered satisfied when the heat transfer ratios (eqs. (38) through (40)) are less than 0.1 percent.

Therefore, it is obvious why the chosen initial design is so critical. Together with the choice of CC it will determine how and to what design the optimizer will converge. As an example, see Table 1. Notice the calculated heat transfer for the initial design. This value is the product of U_m, A, and ΔT_m calculated by ANALIZ using the initial design parameters, some of which are mere estimates. Recall it is the job of the optimizer to vary the design variables in order to bring $\dot{Q}_5$ equal to $\dot{Q}$, $\dot{Q}_3$ and $\dot{Q}_4$ and at the same time minimize the objective function.

In order to remove some of the "artwork" and try to ensure a true optimum design, the following procedure is recommended to generate the design matrix:

(1) Begin with a Type 2 configuration; input the initial design and constraint values enumerated in sections (a) through (d) above; let CC = 10.; execute.

(2) If the heat balance of the resulting design has not converged, but is within 20%, use the design results as a new starting point and set CC = 100 (if the heat balance is not with 20%, let CC = 10.).

TYPE 7 EXCHANGER CONFIGURATION OPTIMUM DESIGN

DESIGN VARIABLES	RUN I		RUN II	
	INITIAL DESIGN	OPTIMUM DESIGN	INITIAL DESIGN	OPTIMUM DESIGN
D_i , inches	.6786	.5392	2.0	1.168
D_o , inches	.7201	.5767	2.5	1.278
ℓ , inches	.1618	.128	.46	.294
t , inches	.0217	.0208	.023	.023
S , inches	.08	.08	.111	.08
P_t , inches	1.044	1.02	4.0	2.557
P_L , inches	1.044	.841	2.125	1.899
h , inches	342.1	238.4	350.	172.9
w , inches	286.6	186.2	490.	253.
Q , BTU/hr	21,988,624.	9,999,925.	20,650,352.	9,999,811.
Volume, ft^3	213.12	72.92	553.2	153.83

Table 1

(3) Repeat step (2) until convergence. NOTE: CC may be adjusted up to 150 when approaching convergence. The design should converge following the use of CC = 100. If too much adjustment of CC is necessary, reaching the optimum from the starting point is unlikely.

(4) Ensure a minimum design by beginning step (1) with a different initial design.

(5) After finding an optimum design for a Type 2 configuration, use that design for the starting point for a Type 3 configuration. This assures a reasonable starting point, probably close to the optimum for Type 3.

(6) Repeat steps (1) through (3) for a Type 3 configuration.

(7) Repeat steps (5) and (6) for the remaining configuration.

e. Design Matrix

The design matrix is presented in Table 2. The optimum design is a Type Four configuration and is shown in Figure 18. Typically, for this case study, when starting far from the final design, COPES/CONMIN would require approximately 1700 calls to ANALIZ to reach an optimum. However when beginning from a reasonable starting point with CC = 100, only 600 calls were needed. Note that each call to ANALIZ requires approximately .06 seconds of CPU time on an IBM 360/67.

2. Case Study Two

a. Problem Formulation

An air heater is to be designed to fit into a space 8' X 24' X 4'. The heat exchanger is to heat 1,000,000 lbm/hr of dry air from 75°F to 130°F. 256,000 lbm/hr of water at 200°F is available. Design the heater so that the required air horsepower is at a minimum.

Table 2. CASE STUDY 1 RESULTS

$\dot{m}_h$ = 133333 lbm/hr | $\dot{m}_a$ = 1190476 lbm/hr | Rectangular Fin Profile

Th_1 = 200 °F | Tc_1 = 95 °F | Aluminum Fins

Th_2 = 125 °F | Tc_2 = 130 °F | Copper Tubes

$\dot{Q}$ = 10,000,000 BTU/hr

	INITIAL DESIGN TYPE 1	3 ROW 3 PASS	2 ROW 1 PASS	3 ROW 1 PASS	4 ROW 1 PASS	2 ROW 2 PASS	4 ROW 2 PASS	4 ROW 4 PASS	5 ROW 5 PASS
D_i, inches	2.0	.48	1.096	.733	.444	.679	.539	.535	.573
D_o, inches	2.5	.523	1.134	.772	.481	.720	.576	.575	.609
ℓ, inches	.46	.0824	.393	.291	.276	.162	.128	.0856	.0733
t, inches	.023	.0157	.0235	.0235	.0217	.0217	.0208	.0172	.0165
s, inches	.111	.08	.08	.08	.08	.08	.08	.08	.08
P_t, inches	4.0	.742	1.922	1.364	1.162	1.044	1.02	.922	1.044
P_L, inches	2.125	.707	1.927	1.413	1.041	1.044	.84	.783	.8
h, inches	350	499.5	224.2	174.0	155.4	342.1	238.4	500.	499.3
w, inches	490.	120.8	373.8	229.1	195.7	285.6	186.2	106.3	99.2
$\dot{Q}$, BTU/hr	20650350	9995582	9999134	10000534	9993329	9998108	9999925	9999968	9999278
Volume, ft^3	553.2	66.1	174.1	88.4	63.8	110.1	72.9	81.3	91.2

b. Design Variable Framework

Assuming constant specific heats,

$$c_{p_w} = 1.0 \quad \text{BTU/lbm-°F}$$

$$c_{p_a} = .24 \quad \text{BTU/lbm-°F}$$

eqs. (12) and (13) will yield the required heat transfer rate and the outlet water temperature.

$$\dot{Q} = \dot{m}\, C_{p_a}\, (T_{c_2} - T_{c_1}) = 13.2 \times 10^6 \quad \text{BTU/hr}$$

$$T_{h_2} = \frac{\dot{m}_h\, C_{p_w}\, T_{h_1} - \dot{Q}}{\dot{m}_h} = 148.44 \text{ °F}$$

Therefore, the design variables are as follows:

$.232 < D_i < 2.325$ in.	$D_i^{\,i} = 2.0$ in.
$.25 < D_o < 2.5$ in.	$D_o^{\,i} = 2.5$ in.
$.0625 \text{ in.} < \ell < \infty$	$\ell^{i} = .46$ in.
$.01 < t < .0235$ in.	$t^{i} = .023$ in.
$.08 < S < .125$ in.	$S^{i} = .111$ in.
$0.0 < P_L < \infty$	$P_L^{\,i} = 2.125$ in.
$0.0 < P_t < \infty$	$P_t^{\,i} = 4.00$ in.

c. Constraint Framework

When designing for minimum horsepower, the optimizer will naturally try to drive the design to a maximum volume in order to reduce airside pressure losses. Therefore, it is reasonable to constrain volume as follows:

$$0 < \text{Volume} < 768 \text{ ft}^3$$

The other constraints are:

$$0.0 < \theta < 1.3$$

$$1.0 < \text{DRATIO} < 2.5$$

$$.018 < \text{TUBTH} < .18 \text{ in.}$$

$$0 < \Delta p_w < .14 \text{ psi}$$

and for the heat balance:

$$\frac{\dot{Q}_5}{\dot{Q}} = 1$$

d. Design Matrix

With the tube and fin materials fixed, as in case one, the matrix is presented in Table 3. The optimum design is a Type 4 configuration, as shown in Figure 19. In case study two when starting far from the optimum, the optimizer called ANALIZ approximately 1900 times. When starting close to the final design, for example using the design for a Type 3 configuration as a starting point for the Type 4 design, COPES/CONMIN only required 596 calls.

Table 3. CASE II DESIGN MATRIX

$\dot{m}_h$ = 256,000 lbm/hr | $\dot{m}_c$ = 1,000,000 lbm/hr | Rectangular Fin Profile

Th_1 = 200 °F | Tc_1 = 75 °F | Aluminum Fins/Copper Tubes

Th_2 = 148.44 °F | Tc_2 = 130 °F | $\dot{Q}$ = 13,200,000 BTU/hr

P_∞ = 14. psi | h = 96 in | w = 288 in

DESIGN VARIABLES	INITIAL DESIGN TYPE 1	1 ROW 1 PASS	2 ROW 1 PASS	3 ROW 1 PASS	4 ROW 1 PASS	2 ROW 2 PASS	3 ROW 3 PASS	4 ROW 2 PASS	4 ROW 4 PASS
D_i, inches	2.0	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	2.14	1.81	COULD NOT PROVIDE REQUIRED $\dot{Q}$	2.31	1.93	2.32
D_o, inches	2.5			2.19	1.85		2.347	1.96	2.355
ℓ , inches	.46			1.21	.90		.385	.82	.12
t , inches	.023			.0235	.0235		.0189	.0234	.0176
S , inches	.111			.08	.08		.08	.085	.08
P_t, inches	4.0			4.90	4.93		3.12	4.93	2.68
P_L, inches	2.125			7.43	7.50		4.41	7.50	4.69
$\dot{Q}$, BTU/hr	4486222			13200986	13200819		13202784	13199363	13205473
FHP	51.96			25.96	22.06		128.6	23.52	395.8

V. CONCLUSIONS

The intent of this investigation was to couple an analysis program with a numerical optimization scheme, COPES/CONMIN, to produce a complete, detailed design program for an air-cooled heat exchanger, (HEDSUP). In addition, the analysis program was to be written such that:

1. the variation in the film coefficients with temperature/length of flow path would be taken into account
2. the surface would be optimized
3. it would be iterative free and thus minimize the CPU time required during an actual trade off study.

The results from test cases using ANALIZ coupled with COPES/CONMIN in its present form were unsatisfactory. Although COPES/CONMIN could optimize the objective function satisfying the inequality constraints, a reliable heat balance could not be obtained. The solution to this problem was the addition of the ALMM option to COPES. In this way, the method of feasible directions, which works best with inequality constraints, was used to satisfy the inequality constraints. The multiplier method, which works best with equality constraints, was used to perform the heat balance.

The results of the case studies show that HEDSUP will yield reliable designs for various design objectives and problems with only some trial and error application of the initial Lagrange multiplier. However, precautions must be taken to overcome the relative minima that plague this design

problem. Table 1 shows vividly the problem of relative minima with one "optimum" design having a volume over 100% greater than the "true" optimum.

The value of numerical optimization in a design problem of this size cannot be overemphasized. For example, in case study 1, the problem is taking place in a nine-dimensional design space and intuition on how an optimized design "should" turn out is quickly lost. Figure 20 helps to illustrate this point. When only varying two design variables, h and w, Figure 20 shows that, when beginning from the initial starting point with a Type 7 configuration, a design satisfying both the equality constraint and the air pressure drop constraint could never be found. However, Table 2 indicates how the numerical optimization routine has varied the other seven design variables in order to shift the constraints and yield an optimized design.

The results show that AHDOP did vary the surface design variables: D_i, D_o, ℓ, t, S, P_t and P_L, in order to produce an optimum heat exchanger. This capability of surface optimization is dependent upon the use of the Briggs-Young and Robinson-Briggs Correlations, eqs. (32) and (34a), respectively. The reliability of the correlations as compared to the "conventional" method is questionable. Actual experimental data for a particular tube and pitch will always be the most useful in predicting pressure drop and film coefficients of across banks of finned tubes. However, the correlations

mentioned above cover the ranges and pitches used in air-cooled heat exchangers, and should therefore be sufficiently accurate in predicting H_o and Δp_a.

VI. RECOMMENDATIONS

In addition to the insight that this investigation has given into the generation of an automated air-cooled heat exchanger design, it has also generated an awareness of this investigation's shortcomings. As mentioned in the review of previous work in this area, each of the optimization methods has its own limitations; none is completely general. Presented herein are recommendations for improving upon and furthering development of HEDSUP.

1. HEDSUP should be expanded to include the capability for two-phase tubeside fluids. Mott, et al. [2] discusses a method involving two-phase tubeside fluids that would be compatible with HEDSUP. The modular design of ANALIZ will aid this effort.

2. Research in the area of numerical optimization using discrete variables would benefit HEDSUP immensely. With discrete variables, the design of the exchanger could be accomplished with "off-the-shelf" materials. At present, the use of the optimizer is restricted to continuous variables with continuous first derivatives. The ability to work with discrete variables would also eliminate the need for the design matrix. The optimizer could optimize for type configuration, fin profile, fin material and tube material.

3. Additional research with ALMM is needed to remove the "artwork" involved with choosing an initial multiplier and

thus increase reliability and hence reduce CPU time. The research should be concentrated in two areas: 1) debugging the new optimizer with AFMM and 2) scaling of design variables.

4. The addition of cost as an objective function would increase the attractiveness of HEDSUP. Mott, et al. [2], and Fontein and Wassink, [18], have presented much useful information in this regard.

5. Mechanical constraints such as tube bursting stress and tube vibrations should be included in HEDSUP.

VII. FIGURES

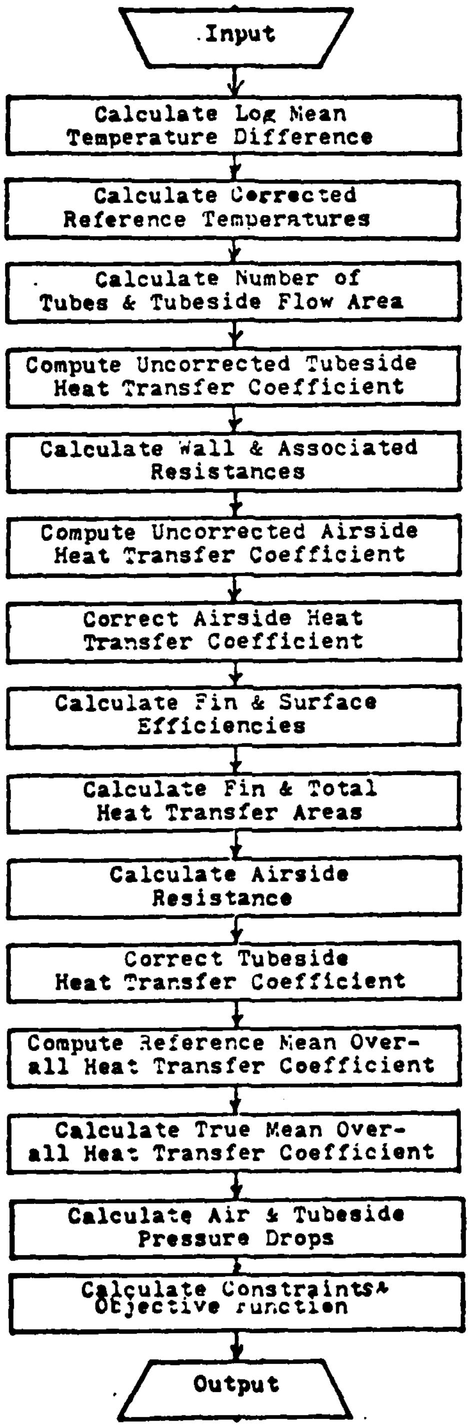
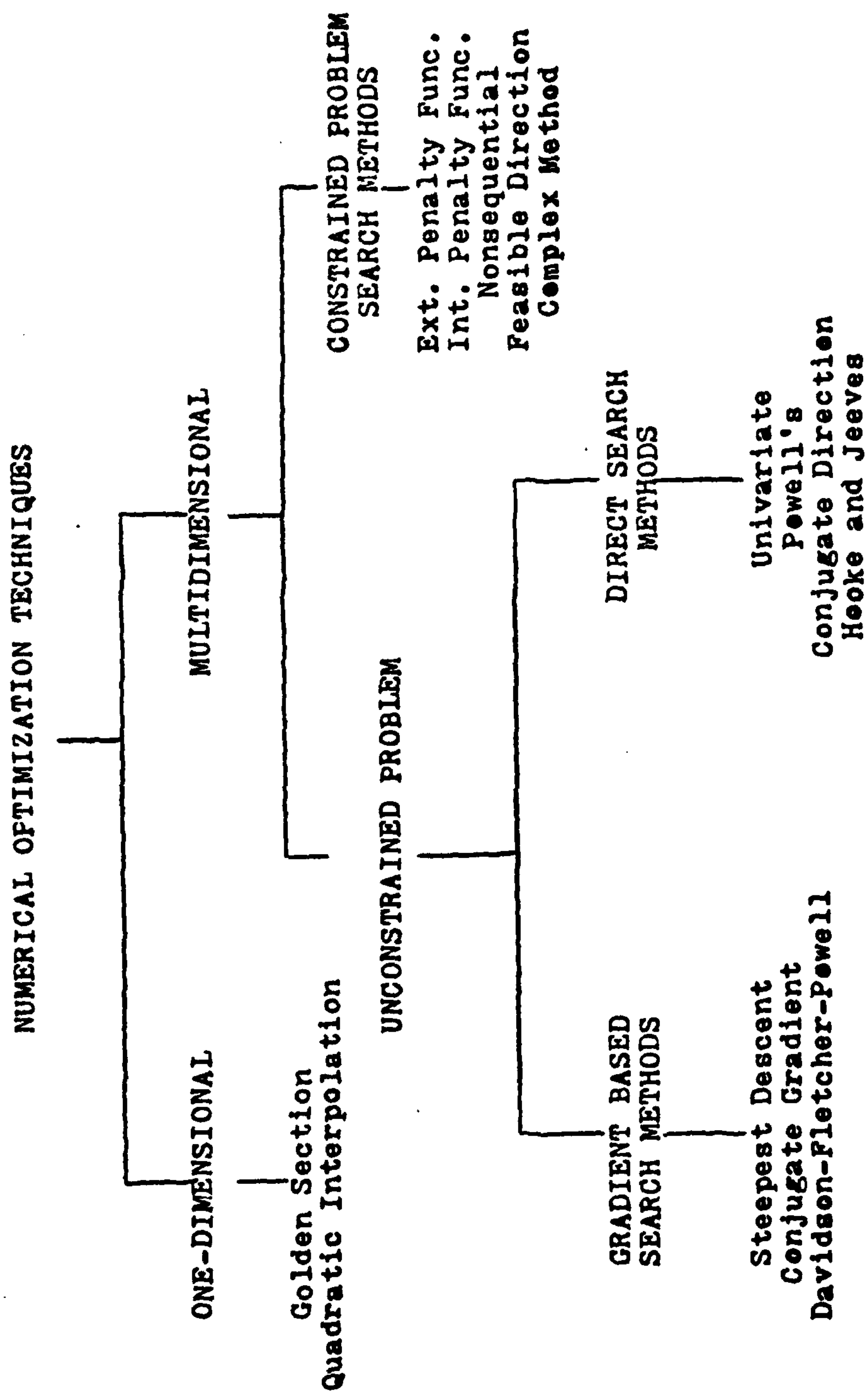


Figure 1



NUMERICAL OPTIMIZATION TECHNIQUES
ONE-DIMENSIONAL
Golden Section
Quadratic Interpolation
MULTIDIMENSIONAL
CONSTRAINED PROBLEM SEARCH METHODS
Ext. Penalty Func.
Int. Penalty Func.
Nonsequential
Feasible Direction
Complex Method
UNCONSTRAINED PROBLEM
DIRECT SEARCH METHODS
Univariate
Powell's
Conjugate Direction
Hooke and Jeeves
GRADIENT BASED SEARCH METHODS
Steepest Descent
Conjugate Gradient
Davidson-Fletcher-Powell

Figure 2

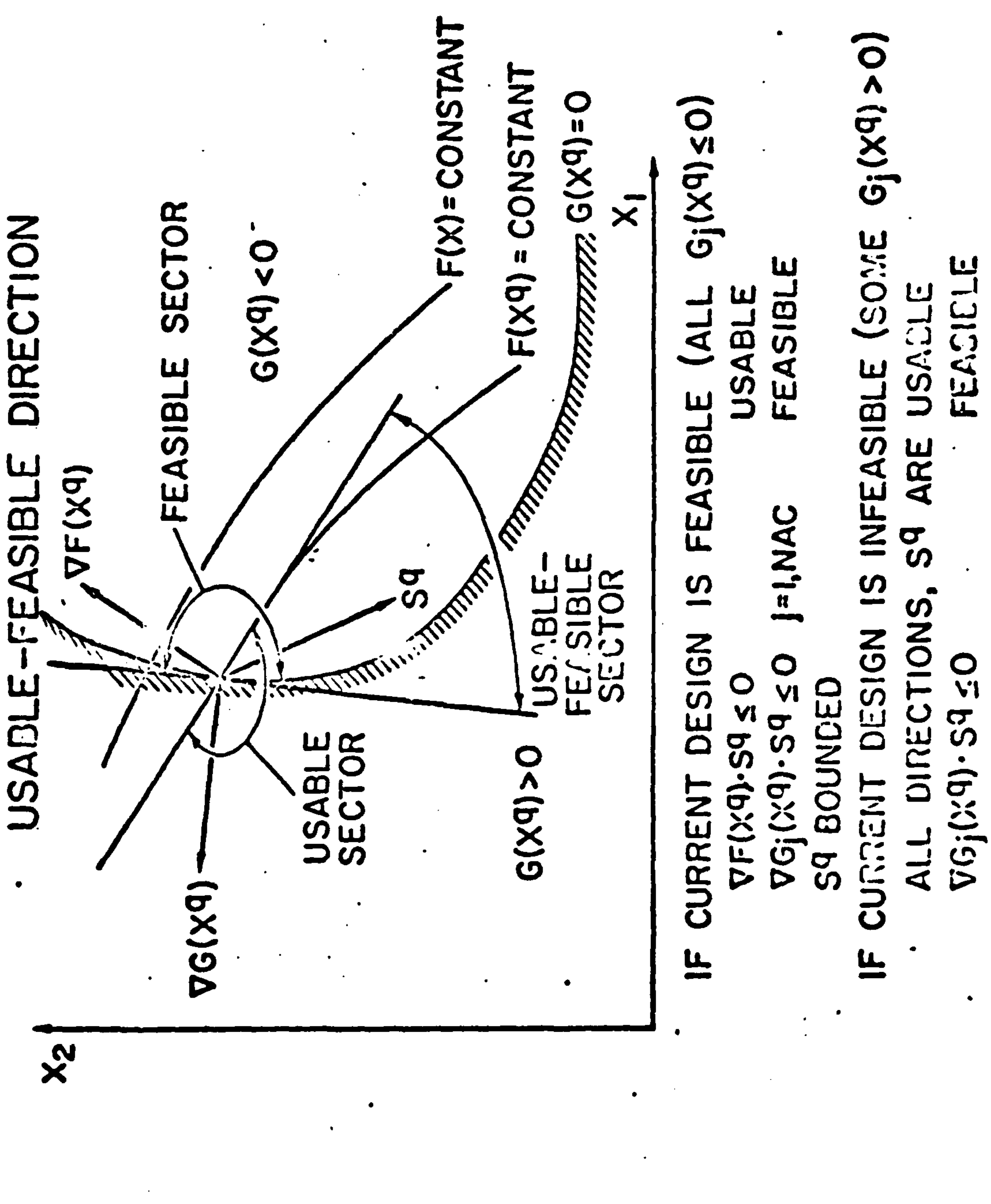

USABLE-FEASIBLE DIRECTION
$\nabla F(X^q)$
FEASIBLE SECTOR
$G(X^q) < 0$
$F(X) = $ CONSTANT
$F(X^q) = $ CONSTANT
$G(X^q) = 0$
X_1
X_2
S^q
USABLE-FEASIBLE SECTOR
USABLE SECTOR
$\nabla G(X^q)$
$G(X^q) > 0$
IF CURRENT DESIGN IS FEASIBLE (ALL $G_j(X^q) \leq 0$)
$\nabla F(X^q) \cdot S^q \leq 0$ USABLE
$\nabla G_j(X^q) \cdot S^q \leq 0$ $j = 1, NAC$ FEASIBLE
S^q BOUNDED
IF CURRENT DESIGN IS INFEASIBLE (SOME $G_j(X^q) > 0$)
ALL DIRECTIONS, S^q ARE USABLE
$\nabla G_j(X^q) \cdot S^q \leq 0$ FEASIBLE

Figure 3

ONE-DIMENSIONAL SEARCH IN DIRECTION S^q

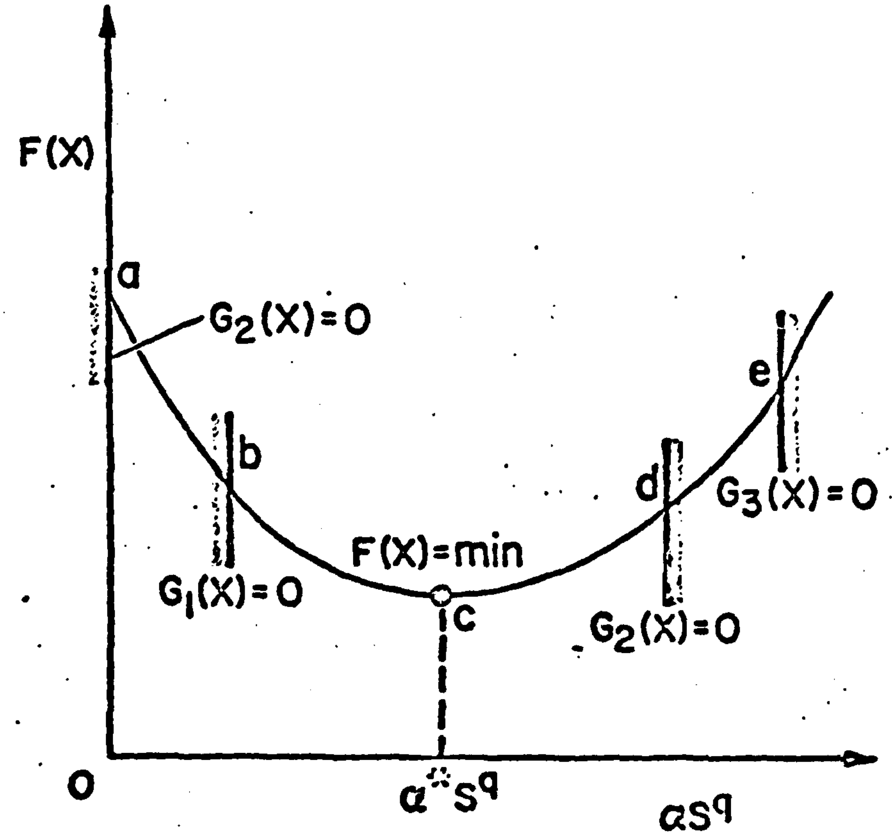
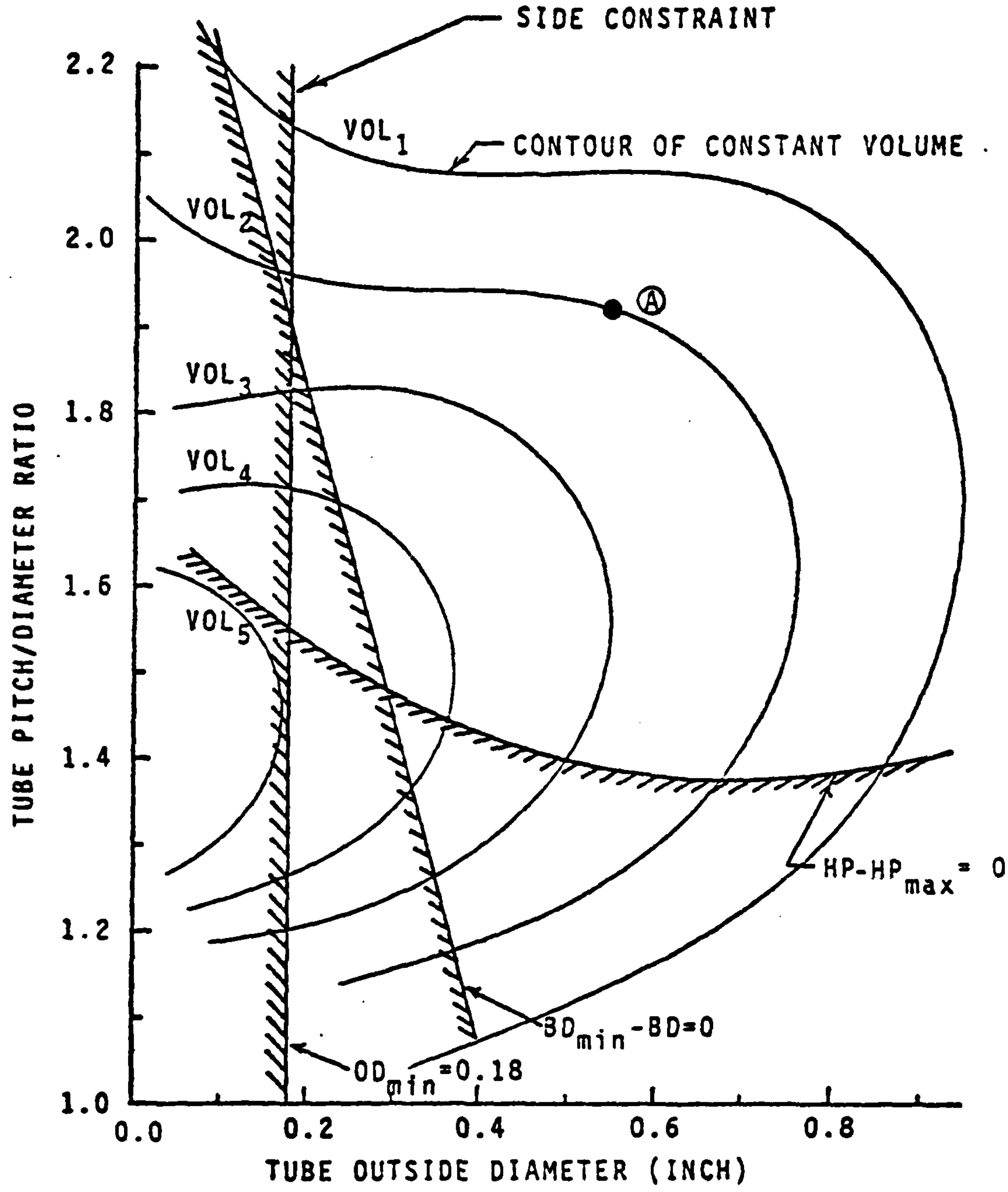
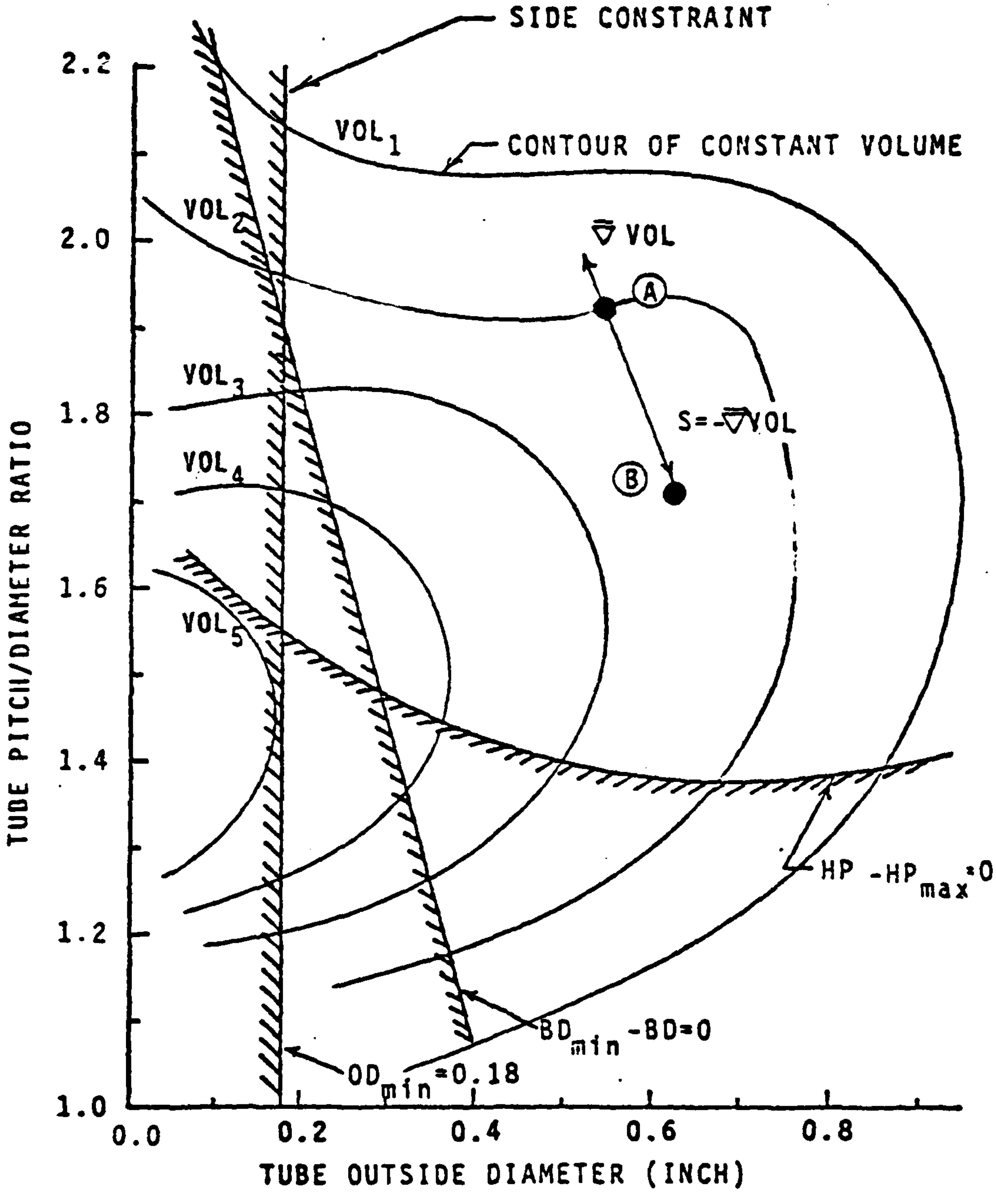


Figure 4



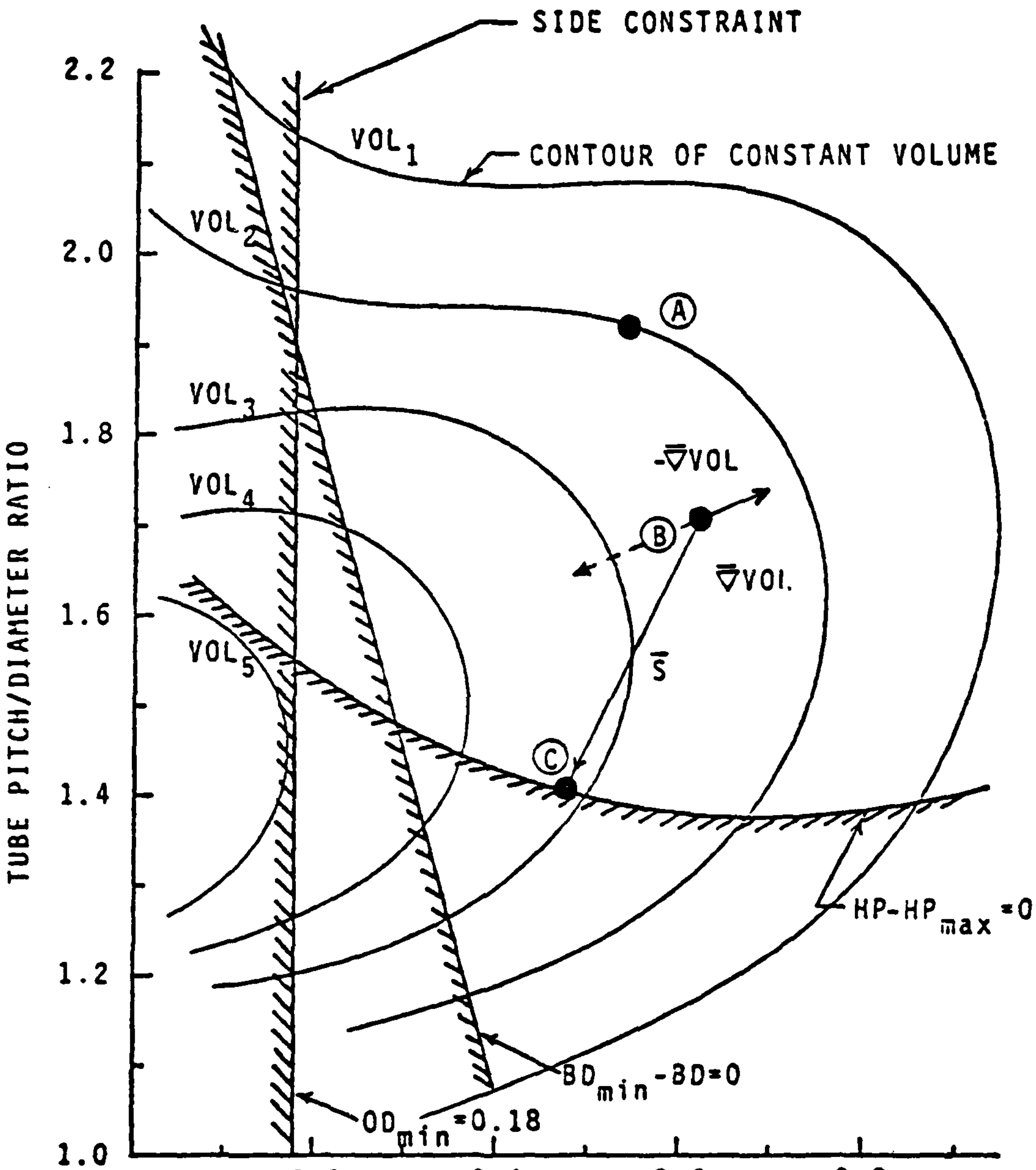
SIDE CONSTRAINT
CONTOUR OF CONSTANT VOLUME
VOL1
VOL2
VOL3
VOL4
VOL5
A
HP-HPmax = 0
BDmin-BD=0
ODmin=0.18
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.0
0.2
0.4
0.6
0.8
TUBE PITCH/DIAMETER RATIO
TUBE OUTSIDE DIAMETER (INCH)

Figure 5



SIDE CONSTRAINT
VOL1
CONTOUR OF CONSTANT VOLUME
VOL2
∇ VOL
A
VOL3
S=-∇VOL
VOL4
B
VOL5
HP -HPmax=0
BDmin-BD=0
ODmin=0.18
TUBE PITCH/DIAMETER RATIO
TUBE OUTSIDE DIAMETER (INCH)
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.0
0.2
0.4
0.6
0.8

Figure 6



SIDE CONSTRAINT
CONTOUR OF CONSTANT VOLUME
VOL1
VOL2
VOL3
VOL4
VOL5
A
B
C
-∇VOL
∇VOL
S
HP-HPmax=0
BDmin-BD=0
ODmin=0.18
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.0
0.2
0.4
0.6
0.8
TUBE PITCH/DIAMETER RATIO
TUBE OUTSIDE DIAMETER (INCH)

Figure 7

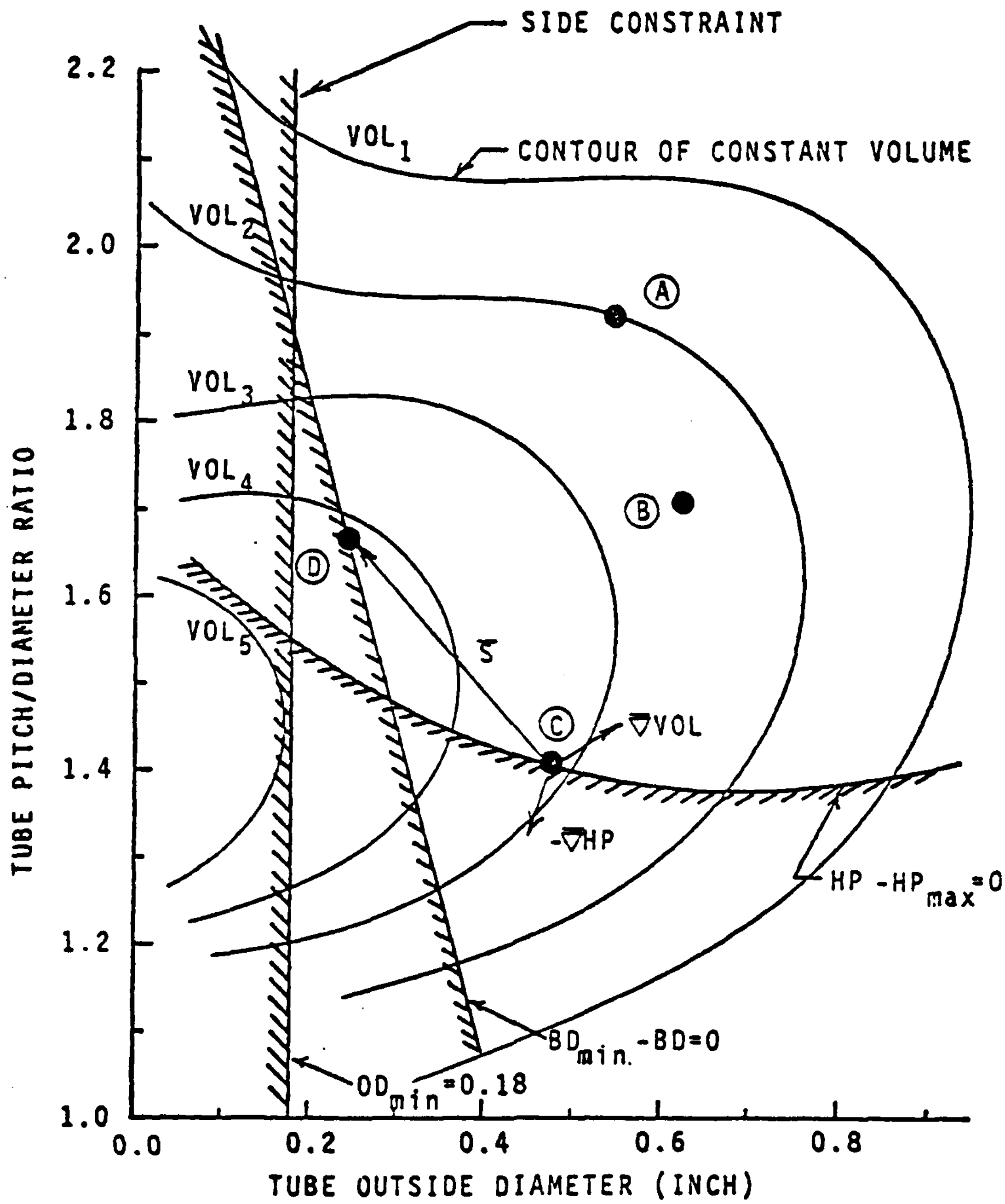
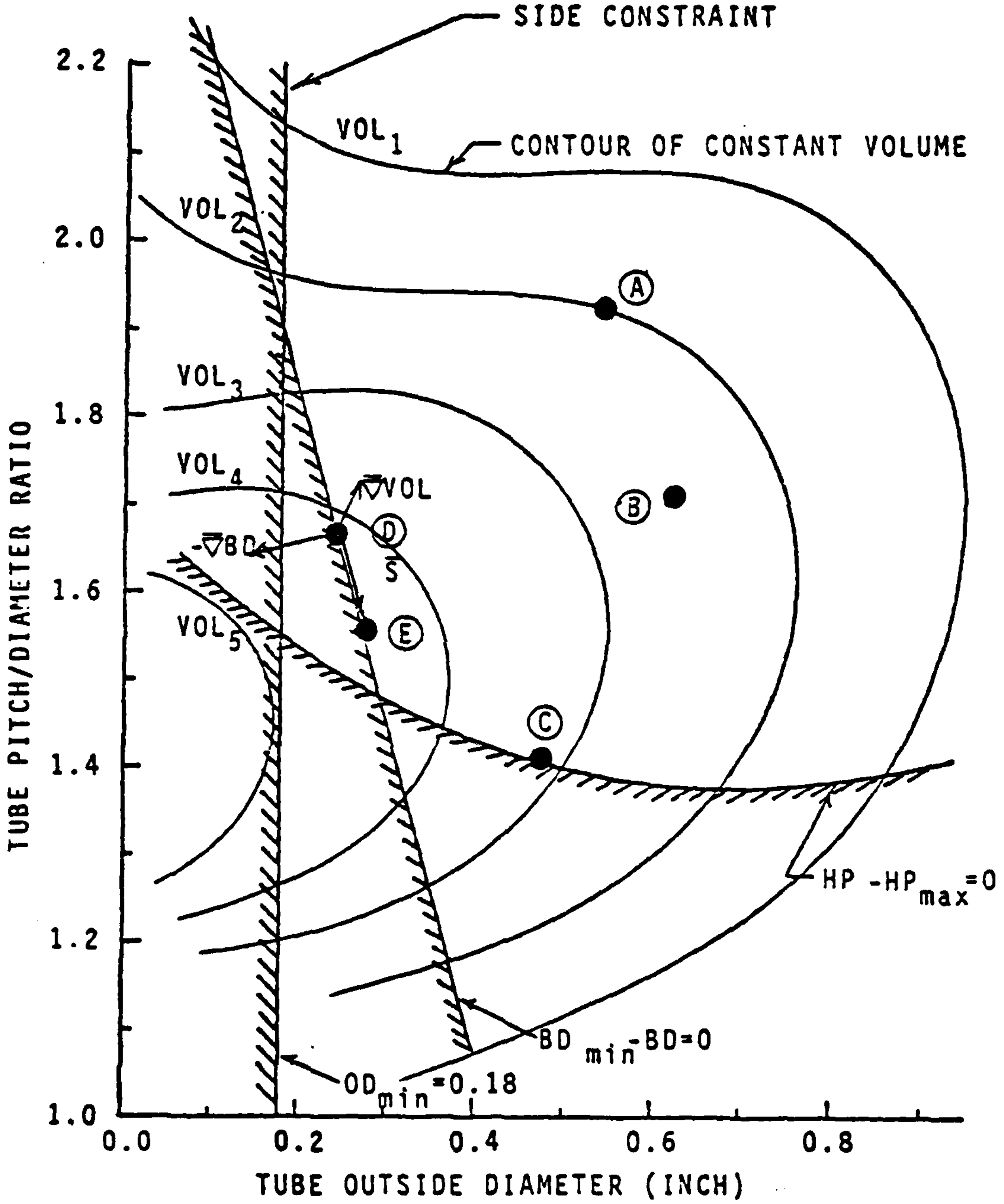

SIDE CONSTRAINT
CONTOUR OF CONSTANT VOLUME
VOL1
VOL2
VOL3
VOL4
VOL5
A
B
C
D
S
∇VOL
-∇HP
HP -HPmax=0
BDmin.-BD=0
ODmin=0.18
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.0
0.2
0.4
0.6
0.8
TUBE PITCH/DIAMETER RATIO
TUBE OUTSIDE DIAMETER (INCH)

Figure 8



SIDE CONSTRAINT
CONTOUR OF CONSTANT VOLUME
VOL_1
VOL_2
VOL_3
VOL_4
VOL_5
A
B
C
D
E
S
$\bar{\nabla}VOL$
$-\bar{\nabla}BD$
$HP - HP_{max} = 0$
$BD_{min} - BD = 0$
$OD_{min} = 0.18$
TUBE PITCH/DIAMETER RATIO
TUBE OUTSIDE DIAMETER (INCH)
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.0
0.2
0.4
0.6
0.8

Figure 9

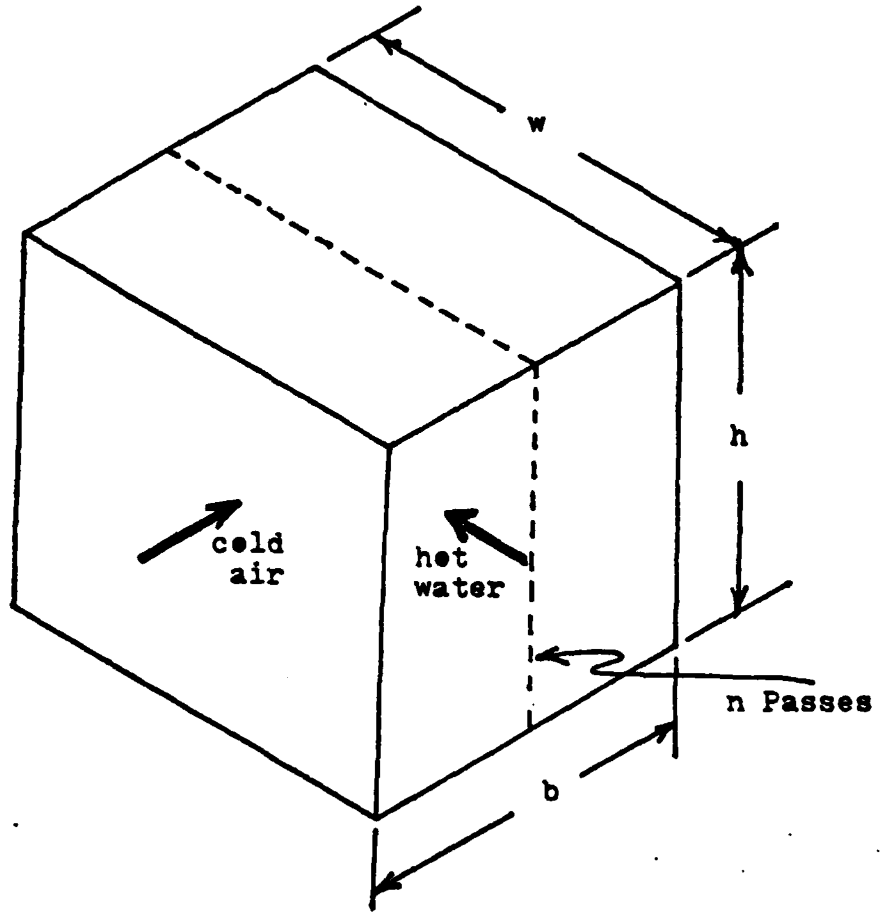


Figure 10. Configuration of Air-Cooled Heat Exchanger

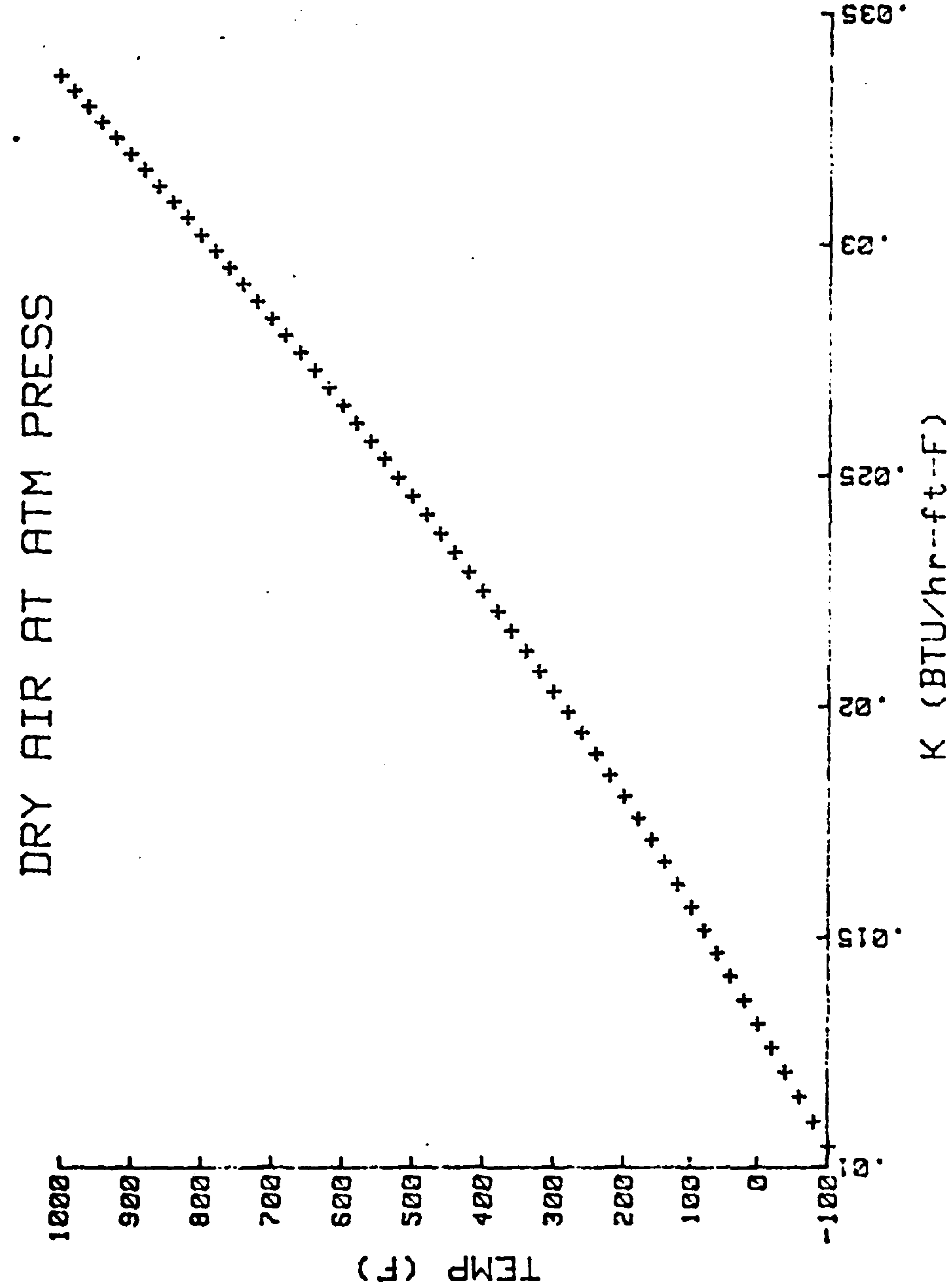

DRY AIR AT ATM PRESS
K (BTU/hr-ft-F)
TEMP (F)
.01
.015
.02
.025
.03
.035
-100
0
100
200
300
400
500
600
700
800
900
1000

Figure 11

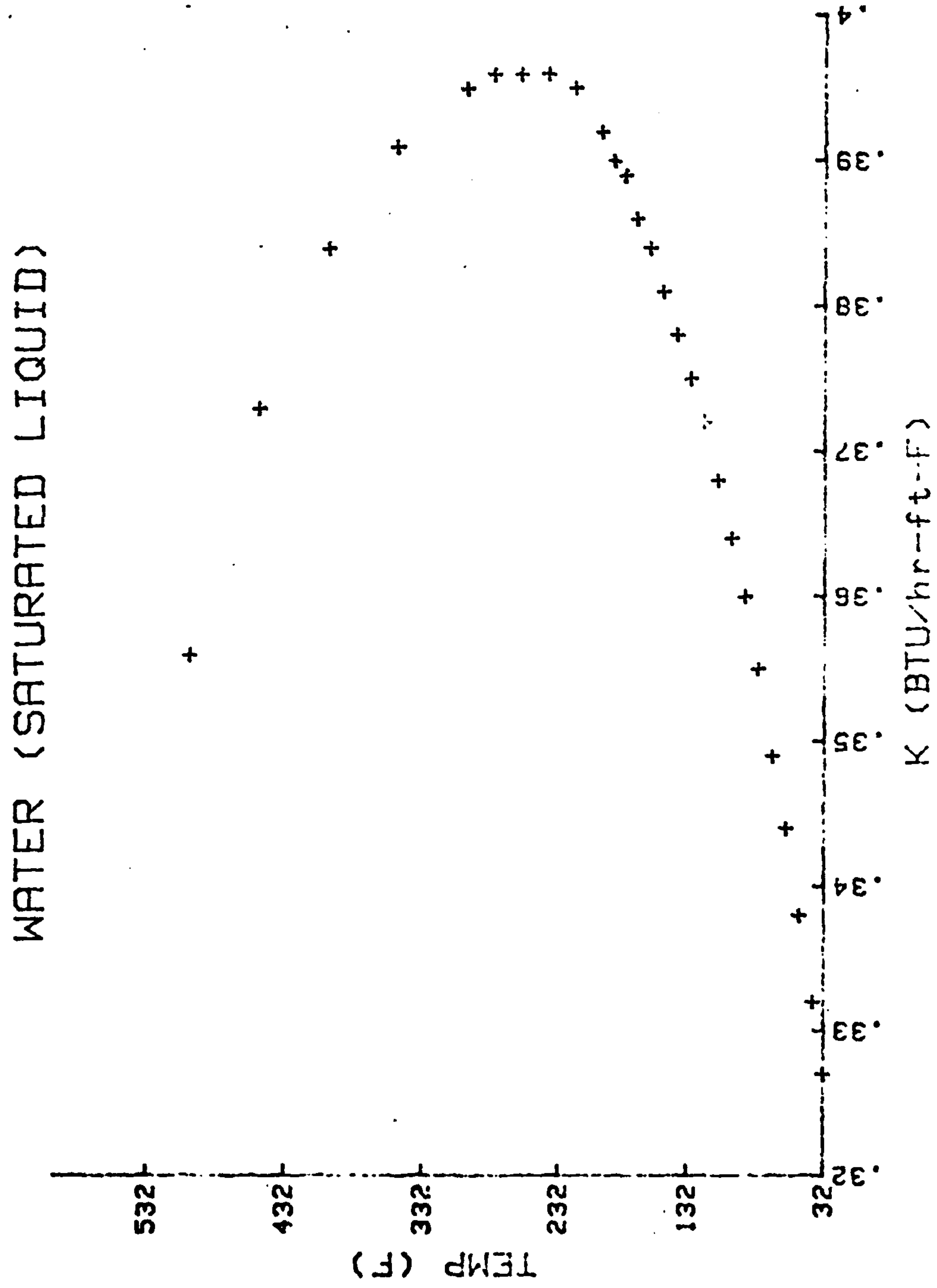

WATER (SATURATED LIQUID)
K (BTU/hr-ft-F)
.32
.33
.34
.35
.36
.37
.38
.39
.4
TEMP (F)
32
132
232
332
432
532

Figure 12

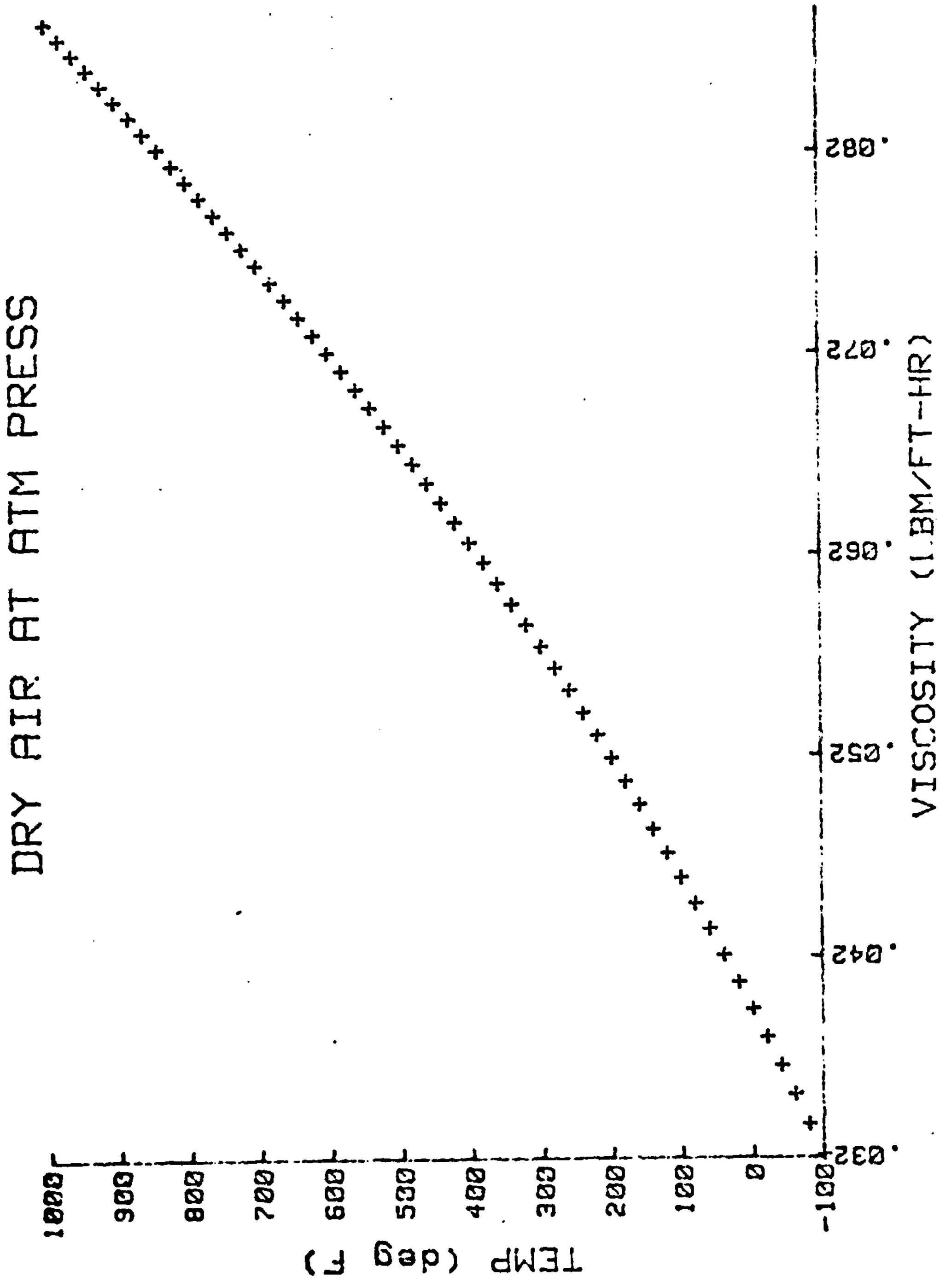

DRY AIR AT ATM PRESS
VISCOSITY (LBM/FT-HR)
.032
.042
.052
.062
.072
.082
TEMP (deg F)
-100
0
100
200
300
400
500
600
700
800
900
1000

Figure 13

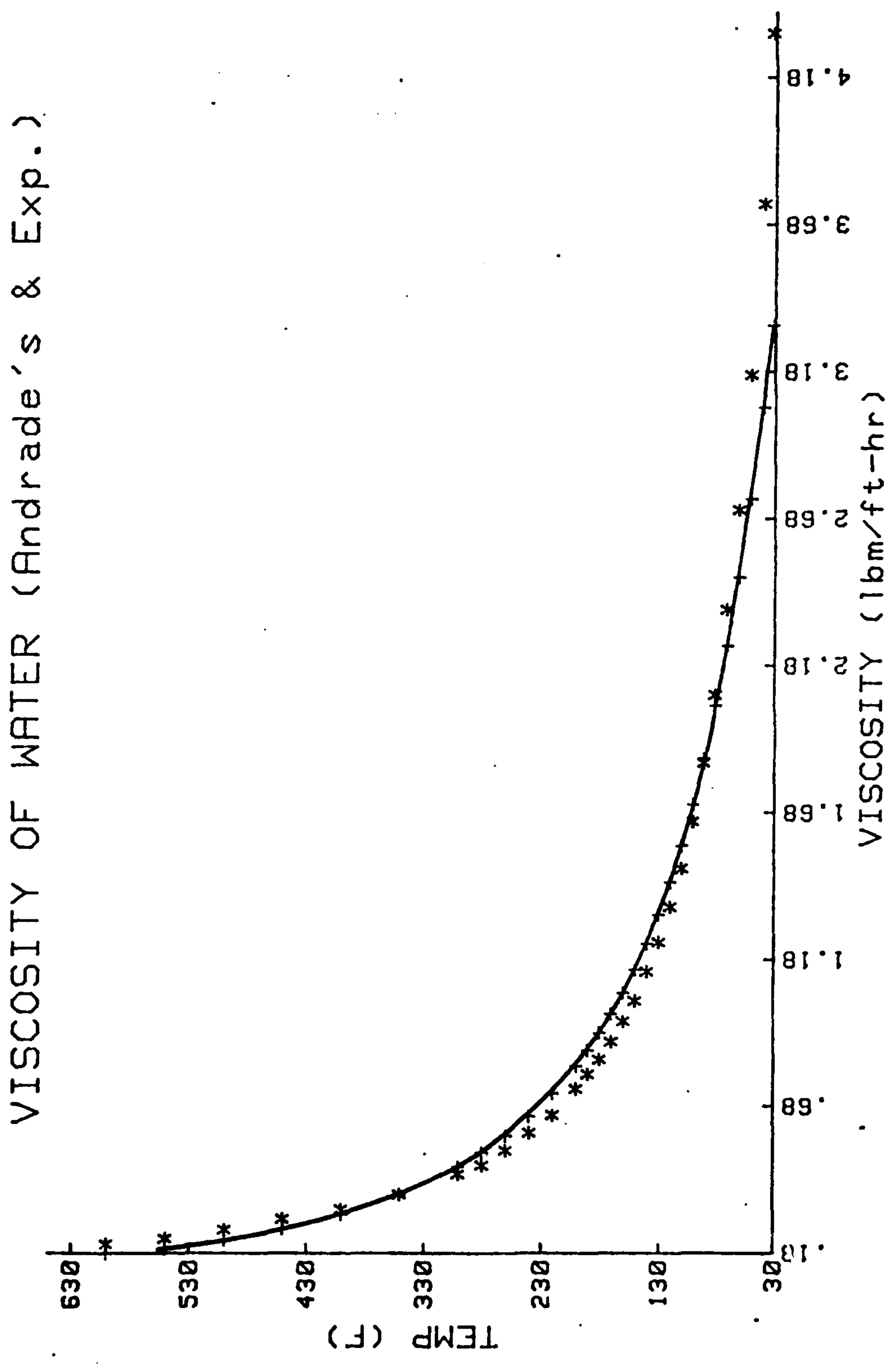

VISCOSITY OF WATER (Andrade's & Exp.)
VISCOSITY (lbm/ft-hr)
TEMP (F)
.18
.68
1.18
1.68
2.18
2.68
3.18
3.68
4.18
30
130
230
330
430
530
630

Figure 14

4 ROW, 2 PASS ARRANGEMENT

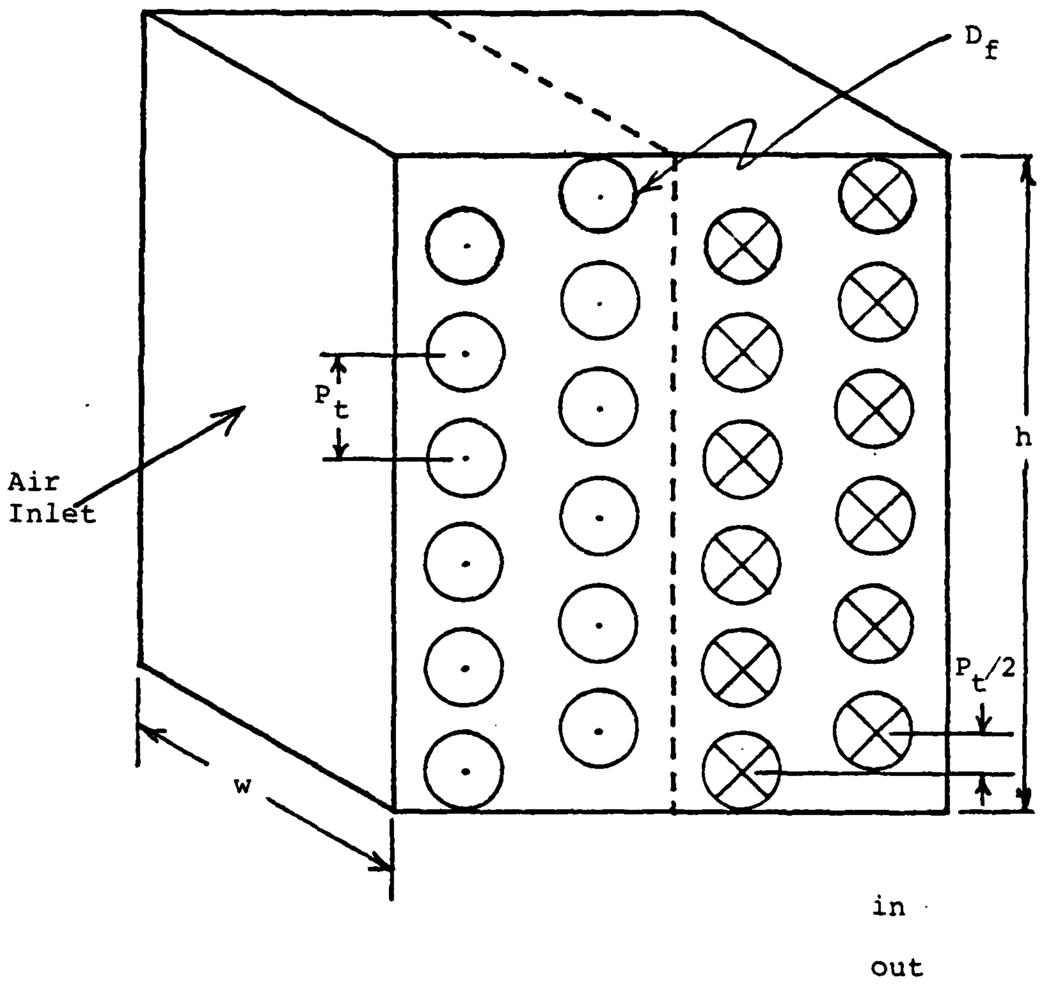


Figure 15

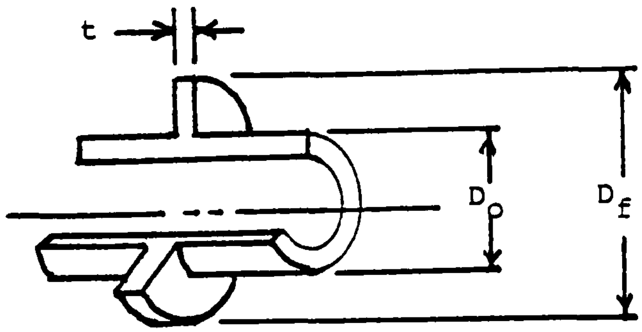


Figure 16

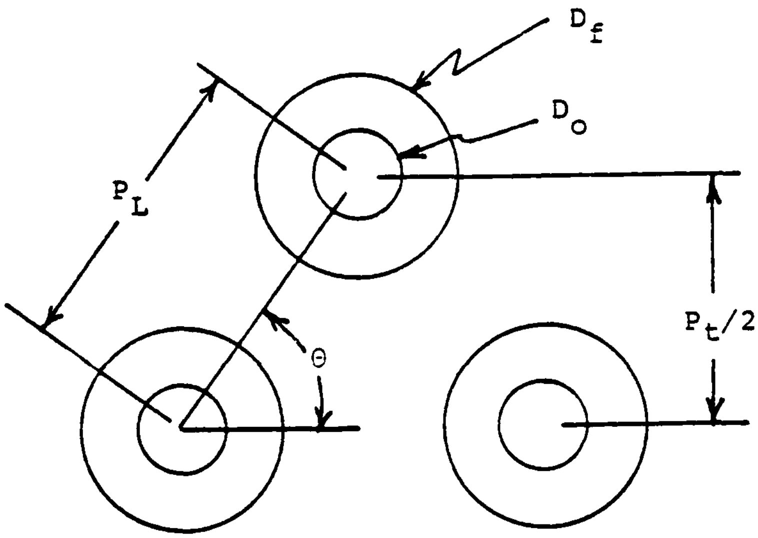
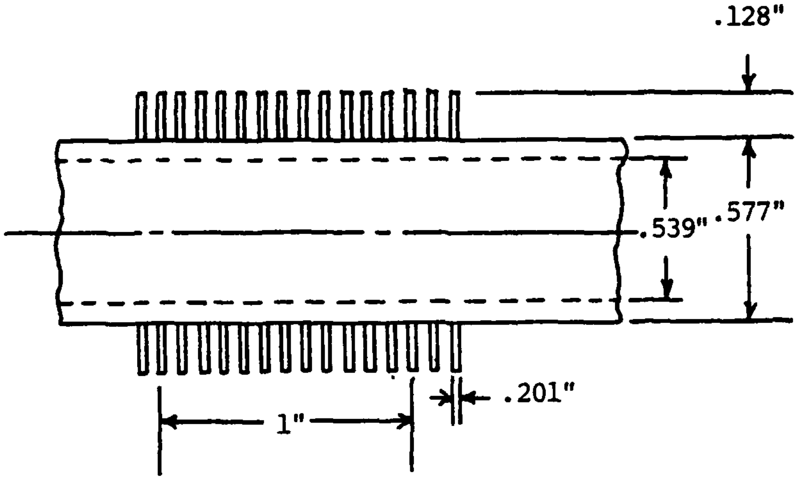


Figure 17

CASE STUDY ONE

DESIGN OPTIMUM



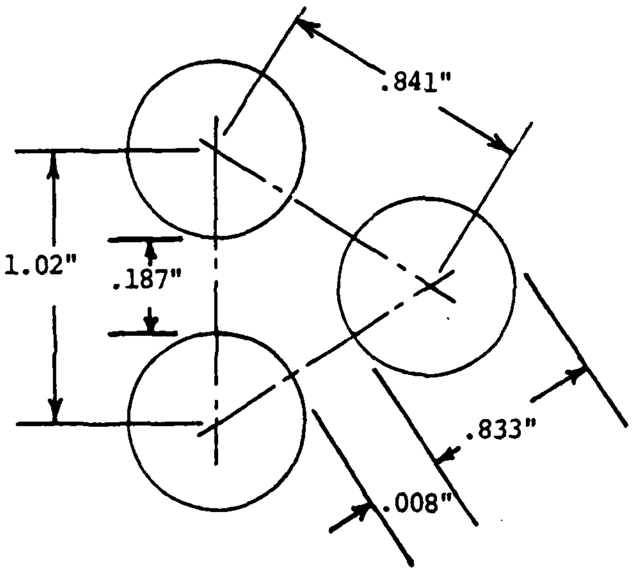
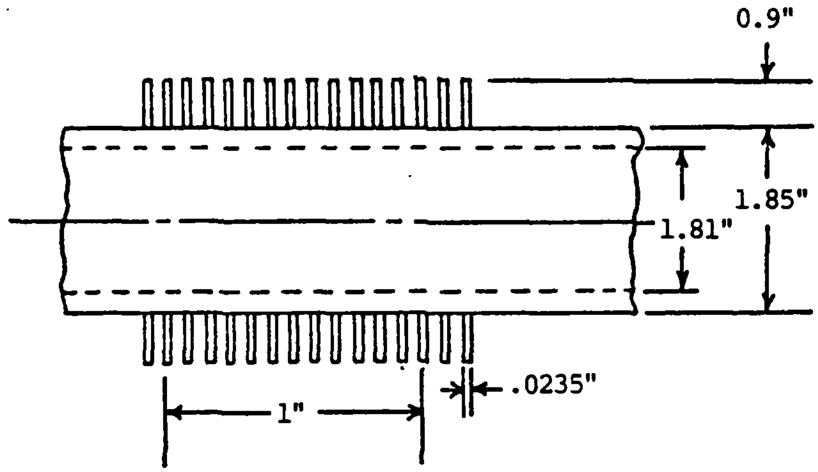


Figure 18

CASE STUDY TWO

OPTIMUM DESIGN



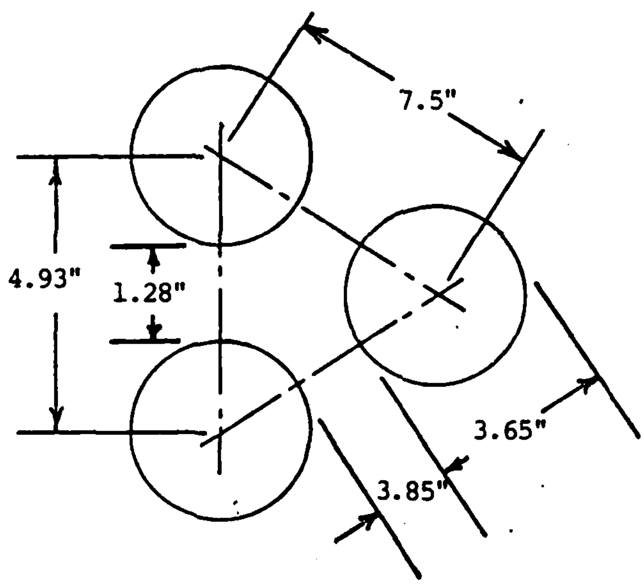


Figure 19

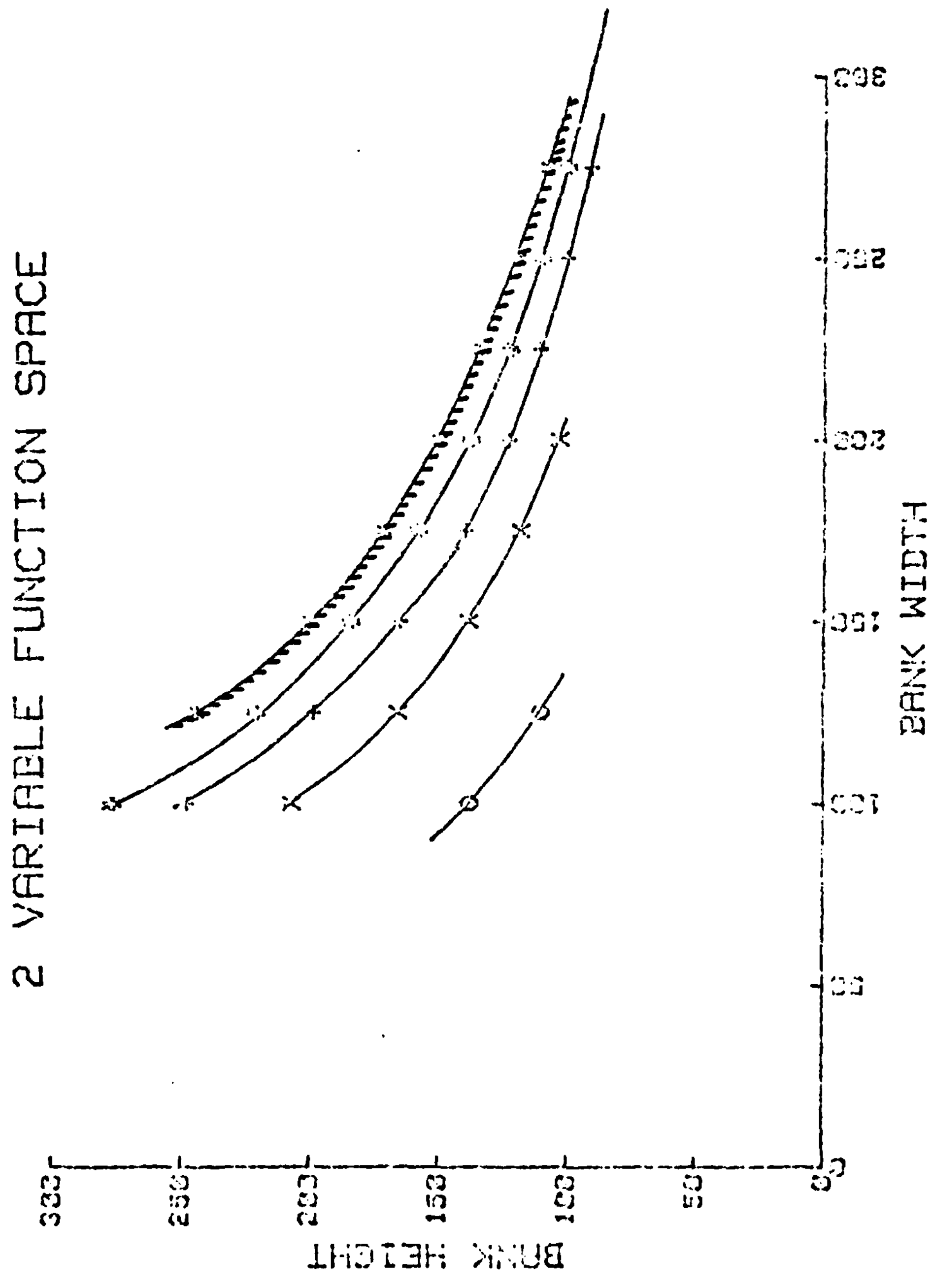

2 VARIABLE FUNCTION SPACE
BANK WIDTH
BANK HEIGHT
0
50
100
150
200
250
300

Figure 20

APPENDIX A

HEAT EXCHANGER DESIGN USING NUMERICAL OPTIMIZATION (HEDSUP)

USERS MANUAL

In order to execute HEDSUP, it is necessary to provide formatted data for COPES [25], followed by formatted data for the ANALIZ portion of the program. This section defines the data which must be supplied. The data is submitted in segmented blocks. All formats are alphanumeric for TITLE and END cards, F10 for real, and I10 for integer data when supplying COPES. For ANALIZ, formats are F14 for real and I10 for integer data.

While the COPES user's sheets define the data in formatted fields of ten, the COPES manual does provide means of simplifying this task through unformatted data input.

The included GLOBAL CATALOG defines objective functions, design variables and constraints along with their location for ease in compiling the necessary COPES data.

GLOBAL CATALOG

GLOBAL LOCATION	FORTRAN NAME	DEFINITION
1	FMDOT	Fluid Mass Flow Rate, lbm/hr
2	FLTMP1	Inlet Fluid Temperature, °F
3	FLTMP2	Outlet Fluid Temperature, °F
4	AMDOT	Air Mass Flow Rate, lbm/hr
5	ARTMP1	Inlet Air Temperature, °F
6	ARTMP2	Outlet Air Temperature, °F
7	TUBEID	Tube Inside Diamter, in
8	TUBEOD	Tube Outside Diameter, in
9	FINHT	Fin Height, in
10	FINTH	Fin Thickness, in
11	FINSP	Fin Spacing, in
12	PITCHL	Longitudinal Pitch, in
13	PITCHN	Transverse Pitch, in
14	BANKW	Bank Width, in
15	BANKH	Bank Height, in
16	VOLUME	Volume, ft^3
17	AREA	Heat Transfer Area, ft^2
18	FHP	Air Horsepower, HP
19	PPA	Airside Pressure Drop, psi
20	PPW	Tubeside Pressure Drop, psi
21	DELQ1	$\dot{Q} - \dot{Q}_5$, BTU/hr
22	DELQ2	$\dot{Q}_5 - \dot{Q}_4$, BTU/hr
23	DELQ3	$\dot{Q}_4 - \dot{Q}_3$, BTU/hr
24	DRATIO	D_f/D_o
25	TUBTH	Tube Wall Thickness, in
26	TOUCHN	Tip-to-Tip Clearance, Transverse
27	TOUCHL	Tip-to-Tip Clearance, Longitudinal

GLOBAL CATALOG (Cont.)

28	PROFH	$T_{c_1} - T_{h_1}$, °F
29	PROFC	$T_{c_1} - T_{h_o}$, °F
30	ARG5	$1 - [2\Delta p_o'/p_\infty]$
31	QDOT1	$\dot{Q}_5$, BTU/hr
32	DELPH1	$\Theta - \theta_{max}$, radians
33	QRATIO	$\dot{Q}_5/\dot{Q}$
34	ARG7	$P_r/2P_L$
35	ARG8	$D_f/2P_L$
36	VROWR	Number of Tubes per Vertical Row
37	DELSFF	Negative of Free Face Area
38	THETA	arcsin (ARG7)

DATA BLOCK A

DESCRIPTION: Title card.

FORMAT AND EXAMPLE

1	2	3	4	5	6	7	8	FORMAT
TITLE								20A4

FIELD	CONTENTS
1-8	Any 80 character title may be given on this card.

DATA BLOCK B

DESCRIPTION: **Program Control Parameters.**

FORMAT AND EXAMPLE

1	2							FORMAT
NCALC	NDV							7I10

FIELD | CONTENTS

1 NCALC: Calculation Control

0 - Read input and stop. Data of blocks A, B and V is required. Remaining data is optional.

1 - One cycle through program. The same as executing ANALIZ stand-alone. Data of blocks A, B and V is required. Remaining data is optional.

2 - Optimization. Data of blocks A-I and V is required. Remaining data is optional.

3 - Augmented Lagrangian Method.

DATA BLOCK C OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Integer optimization control parameters.

FORMAT AND EXAMPLE

1	2	3	4	5	6	7	8	FORMAT
IPRINT	ITMAX	ICNDIR	NSCAL	ITRM	LINOBJ	NACMX1	NFDG	8I10

FIELD — CONTENTS

1 IPRINT: Print control used in the optimization program CONMIN.

- 0 - No print during optimization.
- 1 - Print initial and final optimization information.
- 2 - Print above plus objective function value and design variable values at each iteration.
- 3 - Print above plus constraint values, direction vector and move parameter at each iteration.
- 4 - Print above plus gradient information.
- 5 - Print above plus each proposed design vector, objective function and constraint values during the one-dimensional search.

FIELD	CONTENTS
2	ITMAX: Maximum number of optimization iterations allowed. DEFAULT = 20.
3	ICNDIR: Conjugate direction restart parameter. DEFAULT = NDV + 1.
4	NSCAL: Scaling parameter. GT.0 - Scale design variables to order of magnitude one every NSCAL iterations. LT.0 - Scal design variables according to user-input scaling values.
5	ITRM: Number of consecutive iterations which must satisfy relative or absolute convergence criterion before optimization process is terminated. DEFAULT = 3.
6	LINOBJ: Linear objective function identifier. If the optimization objective is known to be a linear function of the design variables, set LINOBJ = 1. DEFAULT = Non-linear.
7	NACMX1; One plus the maximum number of active constraints anticipated. DEFAULT = NDV + 2.
8	NFDG: Finite difference gradient identifier. 0 - All gradient information is computed by finite difference within CONMIN. 1 - All gradient information is computed analytically by the user-supplied code. 2 - Gradient of objective is computed analytically. Gradients of constraints are computed by finite difference within CONMIN.

REMARKS

1) Currently NFDG must be zero in COPES.

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HEAT EXCHANGER OPTIMIZATION.(U)
SEP 80 C P HEDDERICH

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END
DATE
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4-81
DTIC

DATA BLOCK D OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Floating point optimization program parameters.

FORMAT AND EXAMPLE

1	2	3	4	5	6	7		FORMAT
FDCH	FDCHM	CT	CTMIN	CTL	CTLMIN	THETA		7F10
DELFUN	DABFUN	ALPHAX	ABOBJ1					4F10

NOTE: TWO CARDS ARE READ HERE.

FIELD		CONTENTS
1	FDCH:	Relative change in design variables in calculating finite difference gradients. DEFAULT = 0.01.
2	FDCHM:	Minimum absolute step in finite difference gradient calculations. DEFAULT = 0.001.

FIELD		CONTENTS
3	CT:	Constraint thickness parameter. DEFAULT = -0.05.
4	CTMIN:	Minimum absolute value of CT considered in the optimization process. DEFAULT = 0.004.
5	CTL:	Constraint thickness parameter for linear constraints. DEFAULT = -0.01.
6	CTLMIN:	Minimum absolute value of CTL considered in the optimization process. DEFAULT = 0.001.
7	THETA:	Mean value of push-off factor in the method of feasible directions. DEFAULT = 1.0.
1	DELFUN:	Minimum relative change in objective function to indicate convergence of the optimization process. DEFAULT = 0.001.
2	DABFUN:	Minimum absolute change in objective function to indicate convergence of the optimization process. DEFAULT = 0.001 times the initial objective value.
3	ALPHAX:	Maximum fractional change in any any design variable for first estimate of the step in the one-dimensional search. DEFAULT = 0.1.
4	ABOBJ1:	Expected fractional change in the objective function for first estimate of the step in the one-dimensional search. DEFAULT = 0.1.

REMARKS

1) The DEFAULT values for these parameters usually work well.

DATA BLOCK E OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Total number of design variables, design objective identification and sign.

FORMAT AND EXAMPLE

1	2	3		FORMAT
NDVTOT	IOBJ	SGNOPT		2I10,F10

FIELD		CONTENTS
1	NDVTOT:	Total number of variables linked to the design variables. This option allows two or more parameters to be assigned to a single design variable. The value of each parameter is the value of the design variable times a multiplier, which may be different for each parameter. DEFAULT = NDV.
2	IOBJ:	Global variable location associated with the objective function in optimization.
3	SGNOPT:	Sign used to identify whether function is to be maximized or minimized. +1.0 indicates maximization. -1.0 indicates minimization. If SGNOPT is not unity in magnitude, it acts as a multiplier as well, to scale the magnitude of the objective.

DATA BLOCK F OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Design variable bounds, initial values and scaling factors.

FORMAT AND EXAMPLE

1	2	3	4		FORMAT
VLB	VUB	X	SCAL		4F10

NOTE: READ ONE CARD FOR EACH OF THE NDV INDEPENDENT DESIGN VARIABLES.

FIELD		CONTENTS
1	VLB:	Lower bound on the design variable. If VLB.LT.-1.0E+15, no lower bound.
2	VUB:	Upper bound on the design variable. If VUB.GT.10.E+15, no upper bound.
3	X:	Initial value of the design variable. If X is non-zero, this will supercede the value initialized by the user-supplied subroutine ANALIZ.
4	SCAL:	Design variable scale factor. Not used if NSCAL.GE.0 in BLOCK C.

DATA BLOCK G OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Design variable identification.

FORMAT AND EXAMPLE

1	2	3		FORMAT
NDSGN	IDSGN	AMULT		2I10,F10

NOTE: READ ONE CARD FOR EACH OF THE NDVTOT DESIGN VARIABLES.

FIELD		CONTENTS
1	NDSGN:	Design variable number associated with this variable.
2	IDSGN:	Global variable number associated with this variable.
3	AMULT:	Constant multiplier on this variable. The value of the variable will be the value of the design variable, NDSGN, times AMULT. DEFAULT = 1.0.

DATA BLOCK H OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Number of constrained parameters.

FORMAT AND EXAMPLE

1		FORMAT
NCONS		I10

FIELD	CONTENTS
1	NCONS: Number of constraint sets in the optimization problem.

REMARKS

1) If two or more adjacent parameters in the global common block have the same limits imposed, these are part of the same constraint set.

DATA BLOCK I OMIT IF NDV = 0 IN BLOCK B, OR NCONS = 0 IN BLOCK H

DESCRIPTION: Constraint identification and constraint bounds.

FORMAT AND EXAMPLE

1	2	3	4		FORMAT
ICON	JCON	LCON			3I10
BL	SCAL1	BU	SCAL2		

NOTE: READ TWO CARDS FOR EACH OF THE NCONS CONSTRAINT SETS.
EQUALITY CONSTRAINTS MUST FOLLOW ALL INEQUALITY CONTSTRAINTS.

FIELD		CONTENTS
1	ICON:	First global number corresponding to the constraint set.
2	JCON:	Last global number corresponding to the constraint set. DEFAULT = ICON.
3	LCON:	Linear constraint identifier for this constraint set. LCON = 1 indicates linear constraints, LCON = 2 indicates equality constraint.

FIELD		CONTENTS
1	BL:	Lower bound on the constrained variables. If BL.LT.-1.0E+15, no lower bound.
2	SCAL1:	Normalization factor on lower bound. DEFAULT = MAX of ABS(BL), 0.1.
3	BU:	Upper bound on the constrained variables. If BU.GT.1.0E+15, no upper bound.
4	SCAL2:	Normalization factor on upper bound. DEFAULT = MAX of ABS(BU), 0.1.

REMARKS

1) The normalization factor should usually be defaulted.

2) The constraint functions sent to CONMIN are of the form;

(BL - VALUE)/SCAL1 .LE. 0.0 and (VALUE - BU)/SCAL2 .LE. 0.0.

3) Each constrained parameter is converted to two constraints in CONMIN unless ABS(BL) or ABS(BU) exceeds 1.0E+15, in which case no constraint is created for that bound.

DATA BLOCK V

DESCRIPTION: COPES data 'END' card.

FORMAT AND EXAMPLE

1		FORMAT
END		3A1
END		

FIELD	CONTENTS
1	The word 'END' in columns 1-3.

REMARKS

1) This card **MUST** appear at the end of the COPES data.
2) This ends the COPES input data.
3) Data for the user-supplied routine, ANALIZ, follows this.

ANALIZ DATA

DESCRIPTION: Tubeside Parameters

FORMAT

$\dot{m}_h$ (14)	T_{h_1} (28)	T_{h_2} (42)	C_{p_h} (56)		4F14.4

FIELD

- $\dot{m}_h$ - fluid mass flow rate, lbm/hr
- T_{h_1} - inlet fluid temperature, °F
- T_{h_2} - outlet fluid temperature, °F
- C_p - specific heat, BTU/lbm-°F

REMARKS

1) This begins ANALIZ data input.
2) Input must be formatted.
3) Comment cards are no longer permitted.

DESCRIPTION: Airside Parameters

FORMAT

$\dot{m}_a$ (1–14)	T_{c_1} (15–28)	T_{c_2} (29–42)	C_{p_a} (43–56)	p_∞ (57–70)		5F14.4

FIELD

m_a - air mass flow rate, lbm/hr

T_{c_1} - air inlet temperature, °F

T_{c_2} - outlet air temperature, °F

C_{p_a} - specific heat, BTU/lbm-°F

p_∞ - ambient pressure, psi

DESCRIPTION: Tube Geometry

FORMAT

D_i (14)	D_o (28)	ℓ (42)	t (56)	S (70)		5F14.4

FIELD

D_i - inside tube diameter, inches

D_o - outside root tube diameter, inches

ℓ - fin height, inches

t - fin thickness, inches

S - fin spacing, center-to-center, inches

DESCRIPTION: Tube Arrangement

FORMAT

P_t (14)	P_L (28)	h (42)	w (56)		4F14.4

FIELD

P_t - transverse pitch, inches

P_L - longitudinal pitch, inches

h - bank height, inches

w - bank width, inches

DESCRIPTION: Integer Parameters

FORMAT

14	28	42	56		
ITYPE	JTYPE	NROWS	NPASS		4I10

FIELD

ITYPE - configuration

1 - 1 row, 1 pass

2 - 2 rows, 1 pass

3 - 3 rows, 1 pass

4 - 4 rows, 1 pass

5 - 2 rows, 2 passes

6 - 3 rows, 3 passes

7 - 4 rows, 2 passes

FIELD

8 - 4 rows, 4 passes

9 - pure crossflow

10 - pure counterflow

JTYPE - fin profile

1 - rectangular

NROWS - number of rows

NPASS - number of passes

DESCRIPTION: Miscellaneous Parameters

FORMAT

$\dot{Q}$ (14)	k_{TUBE} (28)	k_{FIN} (42)	R (56)		4F14.4

FIELD

$\dot{Q}$ - given heat transfer rate, BTU/hr

k_{TUBE} - thermal conductivity of tube material, BTU/ft-hr-°F

k_{FIN} - thermal conductivity of fin material, BTU/ft-hr-°F

R - gas constant of air; 53.34 ft-lbf/lbm-°R

DESCRIPTION: Penalty Parameters

FORMAT

R1 (1–14)	R2 (15–28)	R3 (29–42)		3F14.4

NOTE

Set R1 = R2 = R3 = 0 if NCALC = 3.

DESCRIPTION: *Optimization Parameters*

FORMAT

14	28	42	56	70		
Y1	Y2	Y3	Y4	Y5		5F14.4

FIELD

Y1 = 1 if minimizing volume, otherwise Y1 = 0

Y2 = 1 if minimizing area, otherwise Y1 = 0

Y3 = 1 if minimizing air horsepower, otherwise Y1 = 0

Y4 = 1 if minimizing air pressure drop, otherwise Y1 = 0

Y5 = 1 if minimizing tubeside pressure drop, otherwise Y1 = 0

NOTE

If NCALC = 3 let Y1 = Y2 = Y3 = Y4 = Y5 = 0

DESCRIPTION: Augmented Lagrangian Function Method Data

FORMAT

10	20	30		
CC	CMULT	CCMAX		3F10.0

FIELD

CC - initial Lagrangian multiplier

CMULT - Lagrangian multiplication factor

CCMAX - maximum value of Lagrangian multiplier

APPENDIX B

SAMPLE USER'S INPUT

COPES DATA

DATA BLOCK A

TITLE	FORMAT
AIR-COOLED HEAT EXCHANGER DESIGN - CASE I	20A4

DATA BLOCK B

								COMMENT
NCALC	NDV							FORMAT
3	9							7I10

DATA BLOCK C - OMIT IF NDV = 0

								COMMENT
IPRINT	ITMAX	ICNDIR	NSCAL	ITRM	LINOBJ	NACMX1	NFDG	FORMAT
0	0	0	10	0	0	0	0	8I10

DATA BLOCK D - OMIT IF NDV = 0

$								COMMENT
FDCH	FDCHM	CT	CTMIN	CTL	CTLMIN	THETA		FORMAT
0	0	0	0	0	0	0		7F10
DELFUN	DABFUN	ALPHAX	ABOBJ1					FORMAT
0	0	0	0					4F10

DATA BLOCK E - OMIT IF NDV = 0

$				COMMENT
NDVTOT	IOBJ	SGNOBJ		FORMAT
9	16	-1.0		2I10,F10

DATA BLOCK F - OMIT IF NDV = 0

$					COMMENT
VLB	VUB	X	SCAL		FORMAT
.232	2.325	2.0	0.0		4F10
.25	2.5	2.5	0.0		

DATA BLOCK F - CONT.

.0625	1.0 + 20	.460	0.0		
.01	.0235	.023	0.0		
.08	.125	.111	0.0		
0.0	4.0	2.125	0.0		
0.0	4.0	4.0	0.0		
0.0	500.	490.	0.0		
0.0	500.	350.	0.0		

DATA BLOCK G - OMIT IF NDV = 0

$				COMMENT
NDSCN	IDSCN	AMULT		FORMAT
1	7	0.0		2I10,F10
2	8	0.0		
3	9	0.0		
4	10	0.0		
5	11	0.0		
6	12	0.0		
7	13	0.0		
8	14	0.0		
9	15	0.0		

DATA BLOCK H - OMIT IF NDV = 0

$		COMMENT
NCONS		FORMAT
11		I10

DATA BLOCK I - OMIT IF NDV = 0 OR NCONS = 0

$					COMMENT
ICON	JCON	LCON			FORMAT
24	24	0			3I10
$					COMMENT
BL	SCAL1	BU	SCAL2		FORMAT
1.0	0.0	2.5	0.0		4F10
$					
ICON	JCON	LCON			
25	25	0			

DATA BLOCK I - CONT.

$					
BL	SCAL1	BU	SCAL2		
.018	0.0	.18	0.0		
$					
ICON	JCON	LCON			
26	27	0			
$					
BL	SCAL1	BU	SCAL2		
-1.0+20	0.0	0.0	0.0		
$					
ICON	JCON	LCON			
19	19	0			
$					
BL	SCAL1	BU	SCAL2		
-1.0+20	0.0	.0722	0.0		

DATA BLOCK I - CONT.

20	20	0			
$					
BL	SCAL1	BU	SCAL2		
-1.0 + 20	0.0	.14	0.0		
$					
ICON	JCON	LCON			
30	30	0			
$					
BL	SCAL1	BU	SCAL2		
.7	0.0	1.0 + 20	0.0		
$					
ICON	JCON	LCON			
32	32	0			
$					
BL	SCAL1	BU	SCAL2		
-1.0 + 20	0.0	0.0	0.0		

DATA BLOCK I - CONT.

34	35	0			
$					
BL	SCAL1	BU	SCAL2		
-1.0	0.0	1.0	0.0		
$					
ICON	JCON	LCON			
36	36	0			
$					
BL	SCAL1	BU	SCAL2		
2.0	0.0	1.0 + 7	0.0		
$					
ICON	JCON	LCON			
37	37	0			
$					
BL	SCAL1	BU	SCAL2		
-1.0 + 20	0.0	0.0	0.0		

DATA BLOCK I - CONT.

33	33	2			
$					
BL	SCAL1	BU	SCAL2		
1.0	0.1	1.0 + 20	0.0		
$					
ICON	JCON	LCON			
$					
BL	SCAL1	BU	SCAL2		
$					
ICON	JCON	LCON			
$					
BL	SCAL1	BU	SCAL2		

DATA BLOCK V - COPES END OF DATA CARD

END		FORMAT
END		3A1

ANALIZ DATA

$\dot{m}_w$	T_{h_1}	T_{h_2}	C_p					FORMAT
133333.33	200.	125.	1.					4F14.4
$\dot{m}_a$	T_{c_1}	T_{c_2}	C_p	P_∞				FORMAT
1190476.2	95.	130.	.24	14.0				5F14.4
D_i	D_o	ℓ	t	s				FORMAT
2.0	2.5	.46	.023	.111				5F14.4
P_t	P_L	h	w					FORMAT
4.0	2.125	350.	490.					4F14.4
ITYPE	JTYPE	NROWS	NPASS					FORMAT
7	1	4	2					4I10
$\dot{Q}$	k_{TUBE}	k_{FIN}	R					FORMAT
10000000.	220.	118.	53.34					4F14.4

ANALIZ DATA - CONT.

R1	R2	R3						FORMAT
0.0	0.0	0.0						3F14.4
Y1	Y2	Y3	Y4	Y5				FORMAT
0.0	0.0	0.0	0.0	0.0				5F14.4
CC	CMULT	CCMAX						FORMAT
10.0	2.0	1000.						3F10.0
								FORMAT

APPENDIX C

SAMPLE OUTPUT FROM COPES

```
AAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAA

  08/16/80  13.54.51
FILE: SAV       DATA      T1                     NAVAL POSTGRADUATE SCHOOL
1


      CCCCCCC   OOCOOOO   PPPPPPP   EEEEEEE   SSSSSSS
      C         O     O   P     P   E         S
      C         O     O   P     P   E         S
      C         O     O   PPPPPPP   EEEE      SSSSSSS
      C         O     O   P         E               S
      C         O     O   P         E               S
      CCCCCCC   OOCOOOO   P         EEEEEEE   SSSSSSS


             C O N T R O L   P R O G R A M
                          F O R
         E N G I N E E R I N G   S Y N T H E S I S


                        T I T L E
         AIR HEATER DESIGN - OPTIMIZE VOLUME.
1     CARD IMAGES OF CONTROL DATA

      CARD                         IMAGE
0
        1)    AIR HEATER DESIGN - OPTIMIZE VOLUME.
        2)   2,9
        2)              2            9
        3)   $ BLOCK C
        4)   0,80,0,10
        4)              0           80          0          10
        5)   $ BLOCK D
        6)   0.
        6)              0.
        7)   0.
        7)              0.
        8)   $ BLOCK E
        9)   9,16,-1.
        9)              9           16         -1.
       10)   $ BLOCK F
       11)   .232,2.325,.6786
       11)           .232        2.325      .6786
       12)   .25,2.5,.7201
       12)           .25          2.5       .7201
```

```
13)  .0625,1.0+20,.16
13)         .0625     1.0+20         .16
14)  .01,.0217,.0235
14)          .01      .0217        .0235
15)  .08,.125,.08
15)          .08       .125          .08
16)  0.,4.,1.04
16)          0.        4.           1.04
17)  0.,4.,1.04
17)          0.        4.           1.04
18)  0.,500.,285.
18)          0.        500.         285.
19)  0.,500.,342.
19)          0.        500.         342.
20)  $ BLOCK G
21)  1,7
21)           1          7
22)  2,8
22)           2          8
23)  3,9
23)           3          9
24)  4,10
24)           4         10
25)  5,11
25)           5         11
26)  6,12
26)           6         12
27)  7,13
27)           7         13
28)  8,14
28)           8         14
29)  9,15
29)           9         15
30)  $ BLOCK H
31)  11
31)          11
32)  24
32)          24
33)  1.,0.,2.5
33)          1.          0.          2.5
34)  25
34)          25
35)  .018,0.,.18
35)        .018          0.          .18
36)  26,27
36)          26          27
37)  -1.0+20,0.,0.
37)     -1.0+20          0.           0.
38)  19
38)          19
39)  -1.0+20,0.,.0722
39)     -1.0+20          0.        .0722
40)  20
40)          20
41)  -1.0+20,0.,.14
```

```
      41)      -1.0+20          0.         .14
      42)  30
      42)          30
      43)  .7,0.,1.0+20
      43)          .7           0.      1.0+20
      44)  38,38
      44)          38           38
      45)  0.0,0.0,1.3,0.
      45)          0.0          0.0        1.3        0.
      46)  34,35
      46)          34           35
      47)  -1.0,0.,1.
      47)         -1.0          0.         1.
      48)  36
      48)          36
      49)  2.,0.,1.0+7
      49)          2.           0.       1.0+7
      50)  37
      50)          37
      51)  -1.0+20,0.,0.
      51)      -1.0+20          0.         0.
      52)  33,33,2
      52)          33           33         2
      53)  1.,.1,1.0+20
      53)          1.           .1      1.0+20
      54)  END
1    TITLE:
       AIR HEATER DESIGN - OPTIMIZE VOLUME.


     CONTROL PARAMETERS:
     CALCULATION CONTROL,                         NCALC  =     2
     NUMBER OF GLOBAL DESIGN VARIABLES,             NDV  =     9
     INPUT INFORMATION PRINT CODE,              IPNPUT  =     0
     DEBUG PRINT CODE,                            IPDBG  =     0


     CALCULATION CONTROL, NCALC
     VALUE     MEANING
       1       SINGLE ANALYSIS
       2       OPTIMIZATION
       3       SENSITIVITY
       4       TWO-VARIABLE FUNCTION SPACE
       5       OPTIMUM SENSITIVITY
       6       APPROXIMATE OPTIMIZATION


     * * OPTIMIZATION INFORMATION

     GLOBAL VARIABLE NUMBER OF OBJECTIVE              =     16
     MULTIPLIER (NEGATIVE INDICATES MINIMIZATION)     = -0.1000E 01
```

FILE: SAV DATA T1 NAVAL POSTGRADUATE SCHOOL

CONMIN PARAMETERS (IF ZERO, CONMIN DEFAULT WILL OVER-RIDE)

IPRINT	ITMAX	ICNDIR	NSCAL	ITRM	LINOBJ	NACMX1	NFDG
0	80	0	10	0	0	11	0

FDCH	FDCHM	CT	CTMIN
0.0	0.0	0.0	0.0
CTL	**CTLMIN**	**THETA**	**PHI**
0.0	0.0	0.0	0.0
DELFUN	**DABFUN**	**ALPHAX**	**ABOBJ1**
0.0	0.0	0.0	0.0

DESIGN VARIABLE INFORMATION
NON-ZERO INITIAL VALUE WILL OVER-RIDE MODULE INPUT

D. V. NO.	LOWER BOUND	UPPER BOUND	INITIAL VALUE	SCALE
1	0.23200E 00	0.23250E 01	0.67660E 00	0.0
2	0.25000E 00	0.25000E 01	0.72010E 00	0.0
3	0.62500E-01	0.11000E 16	0.16000E 00	0.0
4	0.10000E-01	0.21700E-01	0.23500E-01	0.0
5	0.30000E-01	0.12500E 00	0.80000E-01	0.0
6	0.0	0.40000E 01	0.10400E 01	0.0
7	0.0	0.40000E 01	0.10400E 01	0.0
8	0.0	0.50000E 03	0.28500E 03	0.0
9	0.0	0.50000E 03	0.34200E 03	0.0

DESIGN VARIABLES

ID	D. V. NO.	GLOBAL VAR. NO.	MULTIPLYING FACTOR
1	1	7	0.10000E 01
2	2	8	0.10000E 01
3	3	9	0.10000E 01
4	4	10	0.10000E 01
5	5	11	0.10000E 01
6	6	12	0.10000E 01
7	7	13	0.10000E 01
8	8	14	0.10000E 01
9	9	15	0.10000E 01

CONSTRAINT INFORMATION

THERE ARE 11 CONSTRAINT SETS

ID	GLOBAL VAR. 1	GLOBAL VAR. 2	LINEAR ID	LOWER BOUND	NORMALIZATION FACTOR	UPPER BOUND
1	24	0	0	0.10000E 01	0.10000E 01	0.25000E 01
3	25	0	0	0.13000E-01	0.13000E-01	0.18000E 00
5	26	27	0	-0.11000E 16	0.11000E 16	0.0
7	19	0	0	-0.11000E 16	0.11000E 15	0.72200E-01
8	20	0	0	-0.11000E 16	0.11000E 15	0.14[illegible]E 00
9	30	0	0	0.70000E 00	0.70000E 00	0.11000E 16
10	38	38	0	0.0	0.10000E 00	0.13000E 01

```
     12      34      35      0    -0.10000E 01    0.10000E 01    0.10000E 01
     16      36       0      0     0.20000E 01    0.20000E 01    0.10000E 06
     18      37       0      0    -0.11000E 16    0.11000E 15    0.0
     19      33      33      2     0.10000E 01    0.10000E 00    0.11000E 16

   TOTAL NUMBER OF CONSTRAINED PARAMETERS =    13


   * * ESTIMATED DATA STORAGE REQUIREMENTS

               REAL                               INTEGER
   INPUT  EXECUTION  AVAILABLE       INPUT  EXECUTION  AVAILABLE
    106      558       5000            76      146       1000
1        AIR-COOLED HEAT EXCHANGER OPTIMIZATION


              INPUT DATA


           TUBESIDE PARAMETERS

   MASS FLOW RATE=      133333.3125 LBM/HR
   INLET TEMPERATURE=          200.0000 DEG F
   OUTLET TEMPERATURE=         125.0000 DEG F
   SPECIFIC HEAT=            1.0000 BTU/LBM-F


           AIRSIDE PARAMETERS

   MASS FLOW RATE=     1190476.0000 LBM/HR
   INLET TEMPERATURE=           95.0000 DEG F
   OUTLET TEMPERATURE=         130.0000 DEG F
   SPECIFIC HEAT=            0.2400 BTU/LBM-F
   INLET PRESSURE=          14.0000 PSI


           TUBE GEOMETRY

   TUBE INSIDE DIA.=             0.8700 INCHES
   TUBE OUTSIDE DIA.=             1.0800 INCHES
   FIN HEIGHT=            0.4600 INCHES
   FIN THICKNESS=             0.0190 INCHES
   FIN SPACING, CENTER-TO-CENTER=           0.1110 INCHES


           TUBE ARRANGEMENT
```

FILE: SAV DATA T1 NAVAL POSTGRADUATE SCHOOL

```
TRANSVERSE PITCH=          2.1250 INCHES
LONGITUDINAL PITCH=          2.1250 INCHES
BANK HEIGHT=            96.0000 INCHES
BANK WIDTH=           288.0000 INCHES


     INTEGER PARAMETERS

TYPE OF CROSS FLOW ARRANGEMENT=       7
TYPE OF FIN PROFILE=       1
NUMBER OF ROWS=       4
NUMBER OF PASSES=       2


     MISCELLANEOUS VARIABLES

GIVEN HEAT TRANSFER RATE=   10000000.0000 BTU/HR
THERMAL CONDUCTIVITY OF TUBE MATERIAL=          220.0000 BTU/HR-FT-F
 THERMAL CONDUCTIVITY OF FIN MATERIAL=          118.0000 BTU/HR-FT-F
GAS CONSTANT=            53.3400 FT-LBF/LBM-R


     PENALTY PARAMETERS

  12.000000000            0.0                  0.0


     OPTIMIZATION PARAMETERS

       0.0                0.0                0.0                0.0


   * CC = 0.10000E 02  CMULT = 0.20000E 01  CCMAX = 0.10000E 04
KOUNT=     1


   * * CONMIN DETECTS INITIAL X(I).GT.VUB(I)
   X(I) =  0.2350E-01  VUB(I) =  0.2170E-01
   X(I) IS SET EQUAL TO VUB(I) FOR I =     4

   KCUNT=     1  DEL= 0.77484E 01  OBJ= 0.75310E 02  OBJ1 = 0.72309E 02
-0.43510E 00-0.42596E 00-0.20594E-05-0.90000E 00-0.23558E 01
-0.84404E-01-0.55525E-01-0.69792E-01-0.41434E 00-0.67939E 01
-0.47701E 00-0.10287E 01-0.37127E 00-0.14952E 01-0.50473E 00
-0.10044E 03-0.99998E 00-0.16773E 06 0.77484E 00
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.77484E 01
```

```
 KOUNT=     2

   KOUNT=     2  DEL= 0.22126F 01   OBJ= 0.71452E 02   OBJ1 = 0.72922E 02
-0.45409E 00-0.41835E 00-0.13239E-01-0.39958E 00-0.13120E 01
-0.17861E-03-0.37723E-02-0.42521E-01-0.41393E 00-0.65290E 01
-0.49777E 00-0.16175E 01-0.39251E 00-0.15000E 01-0.50000E 00
-0.11359E 03-0.99998E 00-0.16405E 06-0.22125E 00
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.55358E 01
 KOUNT=     3

   KOUNT=     3  DEL= C.7C381E 00   OBJ= 0.71978E 02   OBJ1 = 0.72243E 02
-0.45477E 00-0.41809E 00-0.33662E-01-0.39663E 00-0.13523E 01
-0.26009E-01-0.15978E-01-0.35173E-02-0.41411E 00-0.65496E 01
-0.47619E 00-0.16091E 01-0.39087E 00-0.14985E 01-0.50155E 00
-0.11315E 03-0.99998E 00-0.16498E 06-0.70381E-01
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.48320E 01
 KOUNT=     4

   KOUNT=     4  DEL= 0.1C330E 01   OBJ= 0.71241E 02   OBJ1 = 0.71464E 02
-0.45449E 00-0.41820E 00-0.19034E-01-0.89814E 00-0.13585E 01
-0.87023E-04-0.32180E-02-0.34043E-02-0.41372E 00-0.65733E 01
-0.49436E 00-0.16110E 01-0.33399E 00-0.15000E 01-0.50001E 00
-0.11309E 03-0.99998E 00-0.16396E 06-0.51651E-01
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.37990E 01
 KOUNT=     5

   KOUNT=     5  DEL= 0.1C048E 01   OBJ= 0.71294E 02   OBJ1 = 0.71459E 02
-0.45451E 00-0.41820E 00-0.16454E-01-0.89835E 00-0.13538E 01
-0.56694E-03-0.33632E-02-0.34971E-02-0.41392E 00-0.65733E 01
-0.49436E 00-0.16110E 01-0.38399E 00-0.15000E 01-0.50003E 00
-0.11309E 03-0.99998E 00-0.16397E 06-0.50240E-01
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.27942E 01
 KOUNT=     6

   KOUNT=     6  DEL= 0.94128E 00   OBJ= 0.71338E 02   OBJ1 = 0.71448F 02
-0.45455E 00-0.41813E 00-0.12723E-01-0.39873E 00-0.13599E 01
-0.17017E-02-0.37034E-02-0.38041E-02-0.41333E 00-0.65733E 01
-0.49436E 00-0.16110E 01-0.36900E 00-0.14999E 01-0.50010E 00
-0.11309E 03-0.99993E 00-0.16399E 06-0.47064E-01
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.0           0.0
 0.0           0.0           0.0           0.18529E 01
```

```
 KOUNT=      7

    KOUNT=      7  DEL= 0.18319E 01  OBJ= 0.71402E 02  OBJ1 = 0.71445E 02
-0.45456E 00-0.41318E 00-0.11355E-01-0.89886E 00-0.18603E 01
-0.21237E-02-0.33754E-02-0.39126E-02-0.41393E 00-0.55733E 01
-0.49436E 00-0.15110E 01-0.38700E 00-0.14999E 01-0.50013E 00
-0.11369E 03-0.99993E 00-0.16400E 06-0.45795E-01
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.21066E-01
 KOUNT=      8

    KOUNT=      8  DEL= 0.17517E 01  OBJ= 0.71483E 02  OBJ1 = 0.71445E 02
-0.45452E 00-0.41819E 00-0.11310E-01-0.89387E 00-0.18606E 01
-0.24354E-02-0.41425E-02-0.39228E-02-0.41394E 00-0.55733E 01
-0.49436E 00-0.15110E 01-0.38700E 00-0.14999E 01-0.50015E 00
-0.11310E 03-0.99993E 00-0.16403E 06-0.43793E-01
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0         -0.17306E 01
 KOUNT=      9

    KOUNT=      9  DEL= 0.16983E 01  OBJ= 0.71555E 02  OBJ1 = 0.71446E 02
-0.45449E 00-0.41820E 00-0.11266E-01-0.89887E 00-0.18608E 01
-0.26459E-02-0.43044E-02-0.39505E-02-0.41394E 00-0.55733E 01
-0.49435E 00-0.15110E 01-0.38700E 00-0.14998E 01-0.50016E 00
-0.11310E 03-0.99998E 00-0.16405E 06-0.42458E-01
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0         -0.34209E 01
 KOUNT=     10

    KOUNT=     10  DEL= 0.25459E 01  OBJ= 0.71530E 02  OBJ1 = 0.71599E 02
-0.45272E 00-0.41371E 00-0.13524E-01-0.89865E 00-0.18589E 01
-0.14077E-01-0.15301E-01-0.55764E-02-0.41411E 00-0.55702E 01
-0.49460E 00-0.15103E 01-0.36924E 00-0.14992E 01-0.50084E 00
-0.11344E 03-0.99998E 00-0.16522E 06 0.31824E-01
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0         -0.88302E 00
 KOUNT=     11

    KOUNT=     11  DEL= 0.25754E 01  OBJ= 0.71609E 02  OBJ1 = 0.71596E 02
-0.45271E 00-0.41391E 00-0.13105E-01-0.89867E 00-0.18589E 01
-0.14149E-01-0.15278E-01-0.55197E-02-0.41411E 00-0.55701E 01
-0.49451E 00-0.15103E 01-0.36925E 00-0.14992E 01-0.50084E 00
-0.11344E 03-0.99998E 00-0.16522E 06 0.32192E-01
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.0          0.0
 0.0          0.0          0.0          0.16924E 01
```

```
 KOUNT=    12

    KOUNT=    12   DEL= 0.25936E 01   OBJ= 0.71691E 02   OBJ1  = 0.71594E 02
 -0.45271E 00-0.41492E 00-0.12367E-01-0.39971E 00-0.13683E 01
 -0.14184E-01-0.15257E-01-0.56143E-02-0.41411E 00-0.65700E 01
 -0.49461E 00-0.15107E 01-0.33925E 00-0.14992E 01-0.50085E 00
 -0.11345E 03-0.99993E 00-0.16522E 06 0.32420E-01
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.42860E 01
 KOUNT=    13

    KCUNT=    13   DEL= 0.17181E 01   OBJ= 0.71739E 02   OBJ1  = 0.71776E C2
 -0.45178E 00-0.41929E 00-0.20604E-01-0.39774E 00-0.13685E 01
 -0.17739E-01-0.13504E-01-0.11582E-01-0.41407E 00-0.65671E 01
 -0.49484E 00-0.16105E 01-0.38949E 00-0.14989E 01-0.50106E 00
 -0.11372E 03-0.99998E 00-0.16497E 06-0.10738E-01
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.25679E 01
 KOUNT=    14

    KCUNT=    14   DEL= 0.16953E 01   OBJ= 0.71757E 02   OBJ1  = 0.71775E C2
 -0.45178E 00-0.41929E 00-0.20463E-01-0.39795E 00-0.13686E 01
 -0.17790E-01-0.13504E-01-0.11582E-01-0.41407E 00-0.65671E 01
 -0.49484E 00-0.16105E 01-0.38949E 00-0.14989E 01-0.50106E 00
 -0.11372E 03-0.99993E 00-0.16497E 06-0.10595E-01
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.87260E 00
 KOUNT=    15

    KCUNT=    15   DEL= 0.16739E 01   OBJ= 0.71774E 02   OBJ1  = 0.71774E C2
 -0.45178E 00-0.41929E 00-0.20317E-01-0.39777E 00-0.13686E 01
 -0.17835E-01-0.13504E-01-0.11582E-01-0.41407E 00-0.65671E 01
 -0.49484E 00-0.16105E 01-0.38949E 00-0.14989E 01-0.50107E 00
 -0.11372E 03-0.99998E 00-0.16497E 06-0.10462E-01
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0           -0.80129E 00
 KOUNT=    16

    KCUNT=    16   DEL= 0.33295E 01   OBJ= 0.71799E 02   OBJ1  = 0.71773E C2
 -0.45178E 00-0.41929E 00-0.20226E-01-0.39798E 00-0.13686E 01
 -0.17860E-01-0.13493E-01-0.11582E-01-0.41407E 00-0.65671E 01
 -0.49434E 00-0.16105E 01-0.38949E 00-0.14989E 01-0.50107E 00
 -0.11371E 03-0.99998E 00-0.16497E 06-0.10405E-01
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0           -0.41308E 01
```

```
 KOUNT=    17

    KOUNT=    17  DEL= 0.21240E 01  OBJ= 0.71803E 02  OBJ1 = 0.71824E 02
 -0.45134E 00-0.41947E 00-0.22213E-01-0.89778E 00-0.13692E 01
 -0.19782E-01-0.16513E-01-0.11521E-01-0.41412E 00-0.55560E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.99998E 00-0.16529E 06 0.66376E-02
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0           -0.20067E 01
 KOUNT=    18

    KOUNT=    18  DEL= 0.21358E 01  OBJ= 0.71817E 02  OBJ1 = 0.71823E 02
 -0.45134E 00-0.41947E 00-0.22132E-01-0.89779E 00-0.13692E 01
 -0.19801E-01-0.16613E-01-0.11519E-01-0.41412E 00-0.55560E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.99993E 00-0.16529E 06 0.66745E-02
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.12912E 00
 KOUNT=    19

    KOUNT=    19  DEL= 0.42976E 01  OBJ= 0.71837E 02  OBJ1 = 0.71822E 02
 -0.45134E 00-0.41946E 00-0.22079E-01-0.89779E 00-0.13692E 01
 -0.19810E-01-0.16502E-01-0.11519E-01-0.41412E 00-0.55560E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.99993E 00-0.16529E 06 0.67151E-02
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.44268E 01
 KOUNT=    20

    KOUNT=    20  DEL= 0.43011E 01  OBJ= 0.71866E 02  OBJ1 = 0.71821E 02
 -0.45134E 00-0.41946E 00-0.22038E-01-0.89780E 00-0.13692E 01
 -0.19803E-01-0.15579E-01-0.11519E-01-0.41412E 00-0.55561E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.99993E 00-0.16529E 06 0.67204E-02
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.0            0.0
  0.0            0.0            0.0            0.87278E 01

    KOUNT=    20  DEL= 0.43011E 01  OBJ= 0.71894E 02  OBJ1 =
1   OPTIMIZATION RESULTS

    OBJECTIVE FUNCTION
    GLOBAL LOCATION    16      FUNCTION VALUE 0.71821E 02

    DESIGN VARIABLES
```

ID	D. V. NO.	GLOBAL VAR. NO.	LOWER BOUND	VALUE	UPPER BOUND
1	1	7	0.23200E 00	0.53830E 00	0.23250E 01
2	2	8	0.25000E 00	0.57559E 00	0.25000E 01
3	3	9	0.62500E-01	0.12989E 00	0.11000E 16
4	4	10	0.99999E-02	0.20777E-01	0.21700E-01
5	5	11	0.79999E-01	0.79999E-01	0.12500E 00
6	6	12	0.0	0.83736E 00	0.39999E 01
7	7	13	0.0	0.10223E 01	0.40000E 01
8	8	14	0.0	0.18607E 03	0.49999E 03
9	9	15	0.0	0.23609E 03	0.49999E 03

DESIGN CONSTRAINTS

ID	GLOBAL VAR. NO.	LOWER BOUND	VALUE	UPPER BOUND
1	24	0.10000E 01	0.14513E 01	0.25000E 01
3	25	0.18000E-01	0.18397E-01	0.18000E 00
5	26	-0.11000E 16	-0.18692E 00	0.0
6	27	-0.11000E 16	-0.19835E-02	0.0
7	19	-0.11000E 16	0.71033E-01	0.72200E-01
8	20	-0.11000E 15	0.13339E 00	0.14000E 00
9	30	0.70000E 00	0.98983E 00	0.11000E 15
10	38	0.0	0.65660E 00	0.13000E 01
12	34	-0.10000E 01	0.61043E 00	0.10000E 01
14	35	-0.10000E 01	0.49582E 00	0.10000E 01
16	36	0.20000E 01	0.22963E 03	0.10000E 03
18	37	-0.11000E 16	-0.16529E 05	0.0
19	33	0.10000E 01	0.99933E 00	0.11000E 16

1

OUTPUT DATA

TUBESIDE PARAMETERS

MASS FLOW RATE= 133333.3125 LBM/HR
INLET TEMPERATURE= 200.0000 DEG F
OUTLET TEMPERATURE= 12[illegible].0000 DEG F
SPECIFIC HEAT= 1.0000 BTU/LBM-F

AIRSIDE PARAMETERS

```
MASS FLOW RATE=     1190476.0000 LBM/HR
INLET TEMPERATURE=         95.0000 DEG F
OUTLET TEMPERATURE=       130.0000 DEG F
SPECIFIC HEAT=           0.2400 BTU/LBM-F
INLET PRESSURE=          14.0000 PSI
```

TUBE GEOMETRY

```
TUBE INSIDE DIA.=            0.5388 INCHES
TUBE OUTSIDE DIA.=             0.5756 INCHES
FIN HEIGHT=            0.1299 INCHES
FIN THICKNESS=           0.0203 INCHES
FIN SPACING, CENTER-TO-CENTER=          0.0800 INCHES
```

TUBE ARRANGEMENT

```
TRANSVERSE PITCH=           1.0223 INCHES
LONGITUDINAL PITCH=            0.8374 INCHES
BANK HEIGHT=          236.0935 INCHES
BANK WIDTH=          186.0704 INCHES

THETA=           0.6556
DELPHI=         -0.3920
```

INTEGER PARAMETERS

```
TYPE OF CROSS FLOW ARRANGEMENT=       7
TYPE OF FIN PROFILE=       1
NUMBER OF ROWS=       4
NUMBER OF PASSES=       2

VERTICAL ROWS CONTAIN        229 TUBES
```

MISCELLANEOUS VARIABLES

```
GIVEN HEAT TRANSFER RATE=  10000000.0000 BTU/HR
THERMAL CONDUCTIVITY OF TUBE MATERIAL=        220.0000 BTU/HR-FT-F
 THERMAL CONDUCTIVITY OF FIN MATERIAL=        118.0000 BTU/HR-FT-F
GAS CONSTANT=          53.3400 FT-LBF/LBM-R
```

PENALTY PARAMETERS

```
 12.000000000        0.0                0.0
```

```
        OPTIMIZATION PARAMETERS
          0.0                0.0                0.0                0.0

        OTHER VALUES

    HEAT TRANSFER RATES
  QDOT1=     9953280.0000 BTU/HR
  QDOT2=     9999998.0000
  QDOT3=     9999994.0000

    HEAT TRANSFER DIFFERENCES
  DELQ1=       6720.0000 BTU/HR
  DELQ2=      -6718.0000
  DELQ3=          4.0000

    PENALTY=          11.9919

    CONSTRAINTS
  DIAMETER RATIO=            1.4513
  TUBE THICKNESS=            0.0184 INCHES
  TOUCHN=           -0.1869
  TOUCHL=           -0.0020
  PROFH=          -70.0000 F
  PROFC=          -30.0000 F

    VALUES TO BE OPTIMIZED
  VOLUME=            71.8214 CU.FT.
  AREA=          10105.4219 SQ.FT.
  AIR HP=            92.7296 HP
  AIR PRESSURE  DROP=            0.0710 PSI
  TUBESIDE PRESSURE DROP=           0.1384 PSI

    OBJECTIVE FUNCTION=          71.8214
1   PROGRAM CALLS TO ANALIZ

       ICALC    CALLS
         1        1
         2     1150
         3        1
```

APPENDIX D

HEDSUP PROGRAM LISTING

```
C                          HEDSUP LISTING
C
C
      SUBROUTINE ANALIZ (ICALC)
C
      DIMENSION A(4,4),CT(2),CTP(2),HI(2),HO(2),EATA(2),SUREFF(2),RESAIR
     1(2),CA(2),DELPAR(2),CHI(2),CHO(2),UK(2),DP(2),DUMMY(2)
C
      COMMON /GLOBCM/ FMDOT,FLTMP1,FLTMP2,AMDOT,ARTMP1,ARTMP2,TUBEID,TUB
     1EOD,FINHT,FINTH,FINSP,PITCHL,PITCHN,BANKW,BANKH,VOLUME,AREA,FHP,PP
     2A,PPW,DELQ1,DELQ2,DELQ3,DRATIO,TUBTH,TOUCHN,TOUCHL,PROFH,PROFC,ARG
     35,QDOT1,DELPHI,QRATIO,ARG7,ARG8,VROWR,DELSFF,QRATO1,QRATO2,THETA
C
      IF (ICALC.GT.1) GO TO 10
C
C ---INPUT INITIAL DATA - WHETHER KNOWN OR ESTIMATED ------------------
C
C TUBESIDE FLUID PROPERTIES:
      READ (5,210) FMDOT,FLTMP1,FLTMP2,CP
C
C  FMDOT - FLUID MASS FLOW RATE, LBM/HR
C
C   FLTMP - INLET FLUID TEMPERATURE, DEG F
C  FLTNP2 - OUTLET FLUID TEMPERATURE, DEG F
C  CP - SPECIFIC HEAT (ASSUMED CONSTANT), BTU/LBM-F
C AIR PROPERTIES:
      READ (5,220) AMDOT,ARTMP1,ARTMP2,ACP,PRES1
C
C  AMDOT - AIR MASS FLOW RATE, LBM/HR
C
C  ARTMP1 - AIR INLET TEMPERATURE, DEG. F
C  ARTMP2 - AIR OUTLET TEMPERATURE, DEG. F
C  ACP - SPECIFIC HEAT (ASSUMED CONSTANT), BTU/LBM-F
C  PRES1 - INLET AIR PRESSURE, PSI
C TUBE GEOMETRY
      READ (5,230) TUBEID,TUBEOD,FINHT,FINTH,FINSP
C
C  TUBEID - TUBE INSIDE DIAMETER, INCHES
C
C  TUBEOD - TUBE OUTSIDE DIAMETER, INCHES
C  FINHT - FIN HEIGHT, INCHES
C  FINTH - FIN THICKNESS, INCHES
C  FINSP - DISTANCE BETWEEN ADJACENT FINS, CENTER-TO-CENTER, INCHES
C TUBE ARRANGEMENT
      READ (5,240) PITCHN,PITCHL,BANKH,BANKW
C
C  PITCHN - DISTANCE BETWEEN CENTERS OF ADLACENT TUBES NORMAL TO FLOW
C           (TRANSVERSE PITCH), INCHES
```

```
C
C  PITCHL - DISTANCE BETWEEN CENTERS OF ADJACENT TUBES PARALLEL TO FLOW
C           MEASURED ALONG THE DIAGONAL (LONGITUDINAL PITCH), INCHES
C
C  BANKH - BANK HEIGHT, INCHES
C
C  BANKW - BANK WIDTH, INCHES
C INTEGER VARIABLES
      READ (5,250) ITYPE,JTYPE,NROWS,NPASS
C
C  ITYPE - TYPE OF CROSS FLOW ARRANGEMENT
C
C  JTYPE - TYPE OF FIN PROFILE
C  NROWS - NO. OF TUBES ROWS IN DIRECTION OF FLOW
C  NPASS - NO. OF TUBE PASSES
C MISCELLANEOUS
      READ (5,260) QDOT,TUBEK,AK,GASCON
C
C  QDOT- GIVEN HEAT TRANSFER RATE, BTU/HR
C
C  TUBEK - THERMAL CONDUCTIVITY OF TUBE MATERIAL, BTU/HR-FT-F
C  AK - THERMAL CONDUCTIVITY OF FIN MATERIAL, BTU/HR-FT-F
C  GASCON - GAS CONSTANT, FT-LBF/LBM-R
C PENALTY PARAMETERS
      READ (5,270) R1,R2,R3
C
C *** INSURE R1= 0 IF QDOT IS NOT A GIVEN QUANTITY ********************
C
C OPTIMIZATION
      READ (5,280) Y1,Y2,Y3,Y4,Y5
C
C SET Y1= 1 IF VOLUME IS TO BE OPTIMIZED, OTHERWISE Y1= 0.
C
C SET Y2= 1 IF HEAT TRANSFER AREA IS TO BE OPTIMIZED, OTHERWISE Y2= 0.
C SET Y3= 1 IF AIR HORSEPOWER IS TO BE OPTIMIZED, OTHERWISE Y3= 0.
C SET Y4= 1 IF AIR PRESSURE DROP IS TO BE OPTIMIZED, OTHERWISE SET Y4= 0
C SET Y5= 1 IF TUBESIDE PRESSURE DROP IS TO BE OPTIMIZED, OTHERWISE Y5=0
C PRINT OUT INPUT
      WRITE (6,110)
C
      WRITE (6,120) FMDOT,FLTMP1,FLTMP2,CP
C
      WRITE (6,130) AMDOT,ARTMP1,ARTMP2,ACP,PRES1
C
      WRITE (6,140) TUBEID,TUBEOD,FINHT,FINTH,FINSP
C
      WRITE (6,150) PITCHN,PITCHL,BANKH,BANKW
C
```

```
      WRITE (6,160) ITYPE,JTYPE,NROWS,NPASS
C
      WRITE (6,170) QDOT,TUBEK,AK,GASCON
C
      WRITE (6,180) R1,R2,R3
C
      WRITE (6,190) Y1,Y2,Y3,Y4,Y5
C
C
C
C
C
C
C
C
C
C
C
C
C
C
C
C
C
C
C
      RETURN
C
C -------EXECUTION----------EXECUTION------------------
C
10    IF (ICALC.GT.2) GO TO 40
C
C ....... MEAN TEMPERATURE DIFFERENCE .........
C
C SUBROUTINE 'LMTD' CALCULATES THE LOG MEAN TEMPERATURE DIFFERENCE OF A
C PURE COUNTERCURRENT EXCHANGER TOGETHER WITH THE MEAN TEMPERATURE
C DIFFERENCE OF THE GIVEN EXCHANGER.
      CALL LMTD (ITYPE,FLTMP1,FLTMP2,ARTMP1,ARTMP2,DELTM,FT,ALMTD,P,Q,R,
     1DIMIM)
C
C
C ........REFERENCE TEMPERATURES ..............................
C
C SUBROUTINE 'REFTEM' CALCULATES THE UNCORRECTED REFERENCE TEMPERATURES
C WHICH WHEN CORRECTED WILL BE USED TO CALCULATE  THE OVERALL HEAT
C TRANSFER COEFFICIENTS IN THE 'USUAL MANNER'.
      CALL REFTEM (FLTMP1,FLTMP2,ARTMP1,ARTMP2,DELT1,DELT2,T1,T1P,T2,T2P
     1,FMDOT,CP,AMDOT,ACP,FLCAP,ARCAP)
```

```
C
C
C ......... CORRECTED TEMPERATURES .............
C
C FOR OTHER THAN PURE COUNTERCURRENT FLOW, THE REFERENCE TEMPERATURES
C MUST BE CORRECTED.
      CALL CORECT (DELTM,ALMTD,T1,T1P,T2,T2P,SI1,SI2,SI1P,SI2P,CT,CTP,FL
     1CAP,ARCAP,DELT1,DELT2)
C
C
C ......CALCULATE NUMBER OF TUBES ..... TUBESIDE FLOW AREA ............
C
C FOR TRIANGULAR PITCH, THE NUMBER OF TUBES WILL BE NEEDED IN ORDER TO
C CALCULATE THE TUBESIDE FLOW AREA.
      CALL TUBES (BANKH,TUBEOD,FINHT,PITCHN,NROWS,NPASS,TUBEID,TAREA,TOT
     1AL,IVROW,FLAREA,VROWR)
C
C
C .............CALCULATE UNCORRECTED INSIDE FILM COEFFICIENT ..........
C
C THE TUBESIDE HEAT TRANSFER COEFFICIENT MUST BE CALCULATED IN THE
C 'USUAL' MANNER FOR BOTH CORRECTED REFERENCE BULK TEMPERATURES. THE
C VISCOSITY CORRECTION FACTOR HAS NOT YET BEEN APPLIED. IN ROETZEL'S
C ITERATION FREE  CALCULATION OF THE WALL TEMPERATURE  DEPENDENT TUBE-
C SIDE FILM COEFFICIENT, ALL HEAT TRANSFER RESISTANCES MUST BE KNOWN
C BEFOREHAND.
      CALL FILMI (CT,TUBEID,FMDOT,FLAREA,L,CP,BANKW,HI)
C
C
C ......... CALCULATE TUBE WALL RESISTANCE .....................
C
      W=(ALOG(TUBEOD/TUBEID))/TUBEK
C
C
C ................ AIRSIDE FILM COEFFICIENT ...................
C
C THE OUTSIDE FILM COEFFICIENT MUST BE CALCULATED IN THE 'USUAL MANNER'
C FOR BOTH CORRECTED REFERENCE TEMPERATURES.
C FIRST, CALCULATE  MAXIMUM VELOCITIES:
      CALL VMAX (ARTMP1,PRES1,GASCON,AMDOT,BANKH,BANKW,FINSP,FINTH,FINHT
     1,TUBEOD,IVROW,STOTAL,SFF,FINPIN,SFIN,SROOT,VMAXF,VMAXS,DELSFF)
C
C
C CALCULATE THE UNCORRECTED AIRSIDE FILM COEFFICIENT BASED ON THE
C CORRECTED REFERENCE TEMPERATURES. THE AIRSIDE FILM COEFFICIENT, HO(I),
C WILL LATER BE CORRECTED FOR THE NUMBER OF ROWS.
C
      CALL FILMO (TUBEOD,AMDOT,SFF,FINSP,FINTH,S,FINHT,CTP,HO,ACP)
```

```
C
C
C CALCULATE THE CORRECTION FACTOR FOR THE AIRSIDE FILM COEFFICIENT
C BASED ON THE NUMBER OF ROWS AS COMPARED TO SIX.
C
      CALL ROWCOR (VMAXS,NROWS,C)
C
C
C THE CORRECT AIRSIDE FILM COEFFICIENTS ARE THE CALCULATED:
C
      CHO(1)=C*HO(1)
C
      CHO(2)=C*HO(2)
C
C
C ................CALCULATE AIRSIDE RESISTANCE ...............
C
C TO CALCULATE THE AIRSIDE RESISTANCE, THE SURFACE EFFICIENCY MUST BE
C COMPUTED. FIRST THE FIN EFFICIENCY FOR BOTH CORRECTED REFERENCE TEM-
C PERATURES IS COMPUTED.
      CALL FINEFF (FINHT,TUBEOD,AK,FINTH,HO,JTYPE,EATA,AM,RO,RE,BETA,ARG
     11,ARG2,B2,B1)
C
C
C NEXT THE FINNED HEAT TRANSFER AND THE TOTAL HEAT TRANSFER AREA ON THE
C AIRSIDE MUST BE CALCULATED.
C
      CALL HTAREA (FINPIN,TUBEOD,FINHT,FINTH,AS,AT,ASAT,TOTAL,BANKW)
C
C
C THE SURFACE EFFICIENCY, SUREFF, IS NOW CALCULATED FOR EACH CORRECTED
C REFERENCE AIRSIDE FILM COEFFICIENT.
C
      SUREFF(1)=1.-ASAT*(1.-EATA(1))
C
      SUREFF(2)=1.-ASAT*(1.-EATA(2))
C
C
C THE AIRSIDE RESISTANCE IS THEREFORE,
C
      RESAIR(1)=1./(SUREFF(1)*CHO(1))
C
      RESAIR(2)=1./(SUREFF(2)*CHO(2))
C
C
C NOW USING ROETZEL'S ITERATION FREE CALCULATION, COMPUTE THE TUBESIDE
C FILM COEFFICIENT CORRECTION:
C
```

```
C ..........CORRECT TUBESIDE FILM COEFFICIENT ...............
      CALL CAY (HI,W,TUBEID,RESAIR,TUBEOD,CT,CTP,CA)
C
C
C NOW CORRECT THE TJBESIDE FILM COEFFICIENT,
C
      CHI(1)=CA(1)*HI(1)
C
      CHI(2)=CA(2)*HI(2)
C
C
C WITH THE CORRECTED FILM COEFFICIENTS CALCULATED AT EACH CORRECTED
C REFERENCE TEMPERATJRE, THE TWO REFERENCE OVERALL HEAT TRANSFER
C COEFFICIENTS BASED ON THE OUTSIDE ROOT TUBE AREA CAN BE COPUTED.
C
      UK(1)=1./((TUBEOD/TUBEID)*(1./CHI(1))+(TUBEOD/24.)*W+RESAIR(1))
C
      UK(2)=1./((TUBEOD/TUBEID)*(1./CHI(2))+(TUBEOD/24.)*W+RESAIR(2))
C
C
C ......... TRUE MEAN OVERALL HEAT TRANSFER COEFFICIENT ...............
C
C THEREFORE, IN ACCORDANCE WITH ROETZEL'S FORMULATION, THE TRUE MEAN
C OVERALL HEAT TRANSFER COEFFICIENT, UKM, IS, IN BTU/HR-SQ.FT.-F :
      UKM=(2.*UK(1)*UK(2))/(UK(1)+UK(2))
C
C
C NOW TO CALCULATE PRESSURE DROP FOR THE AIRSIDE, TAKING INTO ACCOUNT
C THE DENSITY CHANGE.
C
      CALL DELTAP (CTP,TUBEOD,AMDOT,SFF,PITCHN,PITCHL,PRES1,GASCON,NROWS
     1,UK,DELPA,CDELPA,SVOL,ARG5)
C
C
C NOW TO CALCULATE THE TUBESIDE PRESSURE DROP, TAKING INTO ACCOUNT THE
C PROPERTY CHANGES ACROSS THE EXCHANGER.
C
      CALL DELP (CT,TUBEID,FMDOT,FLAREA,CA,UK,DELPW,NPASS,BANKW)
C
C
C THE PARTICULARS HAVE NOW ALL BEEN CALCULATED . THE HEAT BALANCE MUST
C NOW BE PERFORMED ALJNG WITH DEFINING  OBJECTIVE AND CONSTRAINT
C FUNCTIONS.
C
C THE HEAT TRANSFER RATES:
      QDOT1=UKM*AT*DELTM/144.
C
      QDOT2=FLCAP*(FLTMP1-FLTMP2)
```

```
C
      QDOT3=ARCAP*(ARTMP2-ARTMP1)
C
C
C DEFINING SOME HEAT TRANSFER DIFFERENCES FOR USE AS CONSTRAINING
C FUNCTIONS AND AS PENALTIES.
C
      DELQ1=QDOT-QDOT1
C
      QRATIO=QDOT1/QDOT
C
      QRATO1=QDOT2/QDOT1
C
      QRATO2=QDOT3/QDOT2
C
      DELQ2=QDOT1-QDOT2
C
      DELQ3=QDOT2-QDOT3
C
C
C DEFINE THE PENALTY FUNCTION AS:
C
      PENLTY=R1*QRATIO
C
C
C NOW DEFINE CONSTRAINTS WHICH WILL KEEP THE TUBE BANK WITHIN REASONABLE
C LIMITS:
C
C THE RATIO OF FIN DIAMETER TO TUBE DIAMETER NEEDS TO BE KEPT WITHIN
C LIMITS. THE  LOWER LIMIT WILL BE HANDLED BY A SIDE CONSTRAINT ON FINHT
      DRATIO=(2.*FINHT+TUBEOD)/TUBEOD
C
C
C THE TUBE THICKNESS MUST BE KEPT WITHIN REASONABLE  LIMITS:
C
      TUBTH=(TUBEOD-TUBEID)/2.
C
C
C THE TUBES MUST BE KEPT FROM TOUCHING IN BOTH THE LONGITUDINAL AND
C TRANSVERSE DIRECTIONS:
C
      TOUCHN=(TUBEOD+2.*FINHT)-PITCHN
C
      TOUCHL=(TUBEOD+2.*FINHT)-PITCHL
C
C
C THE TEMPERATURE PROFILES IN CROSS FLOW AT BOT ENDS OF THE EXCHANGER
C MUST LOOK REASONABLE.
```

```
C
      PROFH=ARTMP2-=LTMP1
C
      PROFC=ARTMP1-=LTMP2
C
C
C TO PLACE CONSTRAINTS ON THE LONGITUDINAL AND TRANSVERSE PITCH IN ORDER
C TO MAINTAIN AN ISOSOLES TRIANGULAR PITCH.
C
      ARG7=(PITCHN/2.)/PITCHL
C
      ARG8=((TUBEOD+2.*FINHT)/2.)/PITCHL
C
      IF (ARG7.GT.1.) GO TO 20
C
      IF (ARG8.GT.1.) GO TO 20
C
      THETA=ARSIN((PITCHN/2.)/PITCHL)
C
      THETAM=ARCOS(((TUBEOD+2.*FINHT)/2.)/PITCHL)
C
      DELPHI=THETA-THETAM
C
      GO TO 30
C
      THETAM=0.0
C
20    THETA=3.14
C
      DELPHI=3.14
C
C DEFINE THE DIFFERENT OBJECTIVE FUNCTIONS THAT MAY BE USED. THE MUL-
C TIPLIER IN FRONT OF THE PENALTY, Y,  WILL EITHER BE SET TO 1 OR 0
C DEPENDING ON  WHETHER THE FUNCTION IS TO BE OPTIMIZED OR NOT.
C
C EXCHANGER VOLUME IN CUBIC FEET:
30    VOLUM1=(BANKH*BANKW*((TUBEOD+2.*FINHT)+(NROWS-1)*(PITCHL*COS(THETA
     1)))))/1728.
C
      VOLUME=VOLUM1+(Y1*PENLTY)
C
C EXCHANGER AREA IN SQ. FEET:
C
      AREA1=AT/144.
C
      AREA=AT/144.+Y2*PENLTY
C
C AIR HCRSEPOWER:
```

```
C
      FHP1=CDELPA*AMDOT*SVOL*7.27273E-5
C
      FHP=CDELPA*AMDOT*SVOL*7.27273E-5+Y3*PENLTY
C
C PRESSURE DROP IN PSI:
C
      PPA1=CDELPA
C
      PPA=CDELPA+Y4*PENLTY
C
      PPW1=DELPW
C
      PPW=DELPW+Y5*PENLTY
C
C
      RETURN
C
C -------------- PRINT RESULTS --------- PRINT RESULTS ---------------
C
40    WRITE (6,200)
C
      WRITE (6,120) FMDOT,FLTMP1,FLTMP2,CP
C
      WRITE (6,130) AMDOT,ARTMP1,ARTMP2,ACP,PRES1
C
      WRITE (6,140) TUBEID,TUBEOD,FINHT,FINTH,FINSP
C
      WRITE (6,150) PITCHN,PITCHL,BANKH,BANKW
C
      WRITE (6,290) THETA,DELPHI
C
      WRITE (6,160) ITYPE,JTYPE,NROWS,NPASS
C
      WRITE (6,300) IVROW
C
      WRITE (6,170) QDOT,TUBEK,AK,GASCON
C
      WRITE (6,180) R1,R2,R3
C
      WRITE (6,190) Y1,Y2,Y3,Y4,Y5
C
      WRITE (6,320) QDOT1,QDOT2,QDOT3,DELQ1,DELQ2,DELQ3,PENLTY,DRATIO,TU
     1BTH,TOJCHN,TOUCHL,PROFH,PROFC,VOLUM1,AREA1,FHP1,PPA1,PPW1
C
      IF (Y1.EQ.1.) GO TO 50
C
      IF (Y2.EQ.1.) GO TO 60
```

```
C                                                                        A4300
      IF (Y3.EQ.1.) GO TO 70                                             A4310
C                                                                        A4320
      IF (Y4.EQ.1.) GO TO 80                                             A4330
C                                                                        A4340
      IF (Y5.EQ.1.) GO TO 90                                             A4350
C                                                                        A4360
50    WRITE (6,310) VOLUME                                               A4370
C                                                                        A4380
      GO TO 100                                                          A4390
C                                                                        A4400
60    WRITE (6,310) AREA                                                 A4410
C                                                                        A4420
      GO TO 100                                                          A4430
C                                                                        A4440
70    WRITE (6,310) FHP                                                  A4450
C                                                                        A4460
      GO TO 100                                                          A4470
C                                                                        A4480
80    WRITE (6,310) PPA                                                  A4490
C                                                                        A4500
      GO TO 100                                                          A4510
C                                                                        A4520
90    WRITE (6,310) PPW
C                                                                        A4540
C                                                                        A4550
C                                                                        A4560
C                                                                        A4570
C                                                                        A4580
100   RETURN                                                             A4590
C                                                                        A4600
110   FORMAT (1H1,10X,38HAIR-COOLED HEAT EXCHANGER OPTIMIZATION,//////,1  A4610
     15X,10HINPUT DATA)                                                  A4620
120   FORMAT (///,10X,19HTUBESIDE PARAMETERS,//,5X,16HMASS FLOW RATE= ,F  A4630
     114.4,7H LBM/HR,/,5X,19HINLET TEMPERATURE= ,F14.4,6H DEG F,/,5X,20H  A4640
     2OUTLET TEMPERATURE= ,F14.4,6H DEG F,/,5X,15HSPECIFIC HEAT= ,F14.4,  A4650
     310H BTU/LBM-F)                                                     A4660
130   FORMAT (///,10X,18HAIRSIDE PARAMETERS,//,5X,16HMASS FLOW RATE= ,F1  A4670
     14.4,7H LBM/HR,/,5X,19HINLET TEMPERATURE= ,F14.4,6H DEG F,/,5X,20HO  A4680
     2UTLET TEMPERATURE= ,F14.4,6H DEG F,/,5X,15HSPECIFIC HEAT= ,F14.4,1  A4690
     30H BTU/LBM-F,/,5X,16HINLET PRESSURE= ,F14.4,4H PSI)                A4700
140   FORMAT (///,10X,13HTUBE GEOMETRY,//,5X,18HTUBE INSIDE DIA.= ,F14.4  A4710
     1,7H INCHES,/,5X,19HTUBE OUTSIDE DIA.= ,F14.4,7H INCHES,/,5X,12HFIN  A4720
     2 HEIGHT= ,F14.4,7H INCHES,/,5X,15HFIN THICKNESS= ,F14.4,7H INCHES,  A4730
     3/,5X,31HFIN SPACING, CENTER-TO-CENTER= ,F14.4,7H INCHES)           A4740
150   FORMAT (///,10X,16HTUBE ARRANGEMENT,//,5X,18HTRANSVERSE PITCH= ,F1  A4750
     14.4,7H INCHES,/,5X,20HLONGITUDINAL PITCH= ,F14.4,7H INCHES,/,5X,13  A4760
     2HBANK HEIGHT= ,F14.4,7H INCHES,/,5X,12HBANK WIDTH= ,F14.4,7H INCHE  A4770
```

```
      3S)
  160 FORMAT (///,10X,18HINTEGER PARAMETERS,//,5X,32HTYPE OF CROSS FLOW
     1ARRANGEMENT= ,I5,/,5X,21HTYPE OF FIN PROFILE= ,I5,/,5X,16HNUMBER O
     2F ROWS= ,I5,/,5X,18HNUMBER OF PASSES= ,I5)
  170 FORMAT (///,10X,23HMISCELLANEOUS VARIABLES,//,5X,26HGIVEN HEAT TRA
     1NSFER RATE= ,F14.4,7H BTU/HR,/,5X,39HTHERMAL CONDUCTIVITY OF TUBE
     2MATERIAL= ,F14.4,12H BTU/HR-FT-F,/,5X,39H THERMAL CONDUCTIVITY OF
     3FIN MATERIAL= ,F14.4,12H BTU/HR-FT-F,/,5X,14HGAS CONSTANT= ,F14.4,
     413H FT-LBF/LBM-R)
  180 FORMAT (///,10X,18HPENALTY PARAMETERS,//,5X,F14.9,5X,F14.9,5X,F14.
     19)
  190 FORMAT (///,10X,23HOPTIMIZATION PARAMETERS,//,5X,F14.4,5X,F14.4,5X
     1,F14.4,5X,F14.4,5X,F14.4)
  200 FORMAT (1H1,//////////,15X,11HOUTPUT DATA)
  210 FORMAT (4F14.4)
  220 FORMAT (5F14.4)
  230 FORMAT (5F14.4)
  240 FORMAT (4F14.4)
  250 FORMAT (4I10)
  260 FORMAT (4F14.4)
  270 FORMAT (3F14.9)
  280 FORMAT (5F14.4)
  290 FORMAT (/,5X,7HTHETA= ,F14.4,/,5X,8HDELPHI= ,F14.4)
  300 FORMAT (/,5X,21HVERTICAL ROWS CONTAIN,I10,6H TUBES)
  310 FORMAT (/,5X,20HOBJECTIVE FUNCTION= ,F14.4)
  320 FORMAT (///,10X,12HOTHER VALUES,//,5X,19HHEAT TRANSFER RATES,/,3X,
     17HQDOT1= ,F14.4,7H BTU/HR,/,3X,7HQDOT2= ,F14.4,/,3X,7HQDOT3= ,F14.
     24,//,5X,25HHEAT TRANSFER DIFFERENCES,/,3X,7HDELQ1= ,F14.4,7H BTU/H
     3R,/,3X,7HDELQ2= ,F14.4,/,3X,7HDELQ3= ,F14.4,//,5X,9HPENALTY= ,F14.
     44,//,5X,11HCONSTRAINTS,/,3X,16HDIAMETER RATIO= ,F14.4,/,3X,16HTUBE
     5 THICKNESS= ,F14.4,7H INCHES,/,3X,8HTOUCHN= ,F14.4,/,3X,8HTOUCHL=
     6,F14.4,/,3X,7HPROFH= ,F14.4,2H F,/,3X,7HPROFC= ,F14.4,2H F,//,5X,2
     72HVALUES TO BE OPTIMIZED,/,3X,8HVOLUME= ,F14.4,7H CU.FT.,/,3X,6HAR
     8EA= ,F14.4,7H SQ.FT.,/,3X,8HAIR HP= ,F14.4,3H HP,/,3X,20HAIR PRESS
     9URE  DROP= ,F14.4,4H PSI,/,3X,24HTUBESIDE PRESSURE DROP= ,F14.4,4H
     $ PSI)
      END
```

```
      FUNCTION VISCFL (T)
C
C CALCULATES THE VISCOSITY OF WATER IN LBM/FT-HR GIVEN TEMPERATURE IN
C DEGREES F ACCORDING TO ANDRADE'S LAW.
C
      VISCFL=.01339*EXP(2715.7764/(T+460.))
C
      RETURN
      END
```

```
      FUNCTION TCFL (T)
C
C CALCULATES THE THERMAL CONDUCTIVITY OF WATER IN BTU/HR-FT-F GIVEN
C TEMPERATURE IN DEGREES F.
C
      TCFL=.31128+5.84054E-4*T-9.931E-7*T*T
C
      RETURN
      END
```

```
      FUNCTION VISCAR (T)
C
C CALCULATES THE VISCOSITY OF AIR IN LBM/FT-HR GIVEN TEMPERATURE IN
C DEGREES F.
C
      VISCAR=.03939+6.72E-5*T-2.1E-8*T*T
C
      RETURN
      END
```

```
      FUNCTION TCAR (T)
C
C CALCULATES THE THERMAL CONDUCTIVITY OF AIR IN BTU/HR-FT-F GIVEN
C TEMPERATURE IN DEGREES F.
C
      TCAR=.01312+2.62806E-5*T-7.0E-9*T*T
C
      RETURN
      END
```

```
      FUNCTION FF (REYN)
C
C CALCULATES THE TUBESIDE FRICTION FACTOR ACCORDING TO THE STANDARDS OF
C THE TUBULAR EXCHANGER MANUFACTURER'S ASSOCIATION, FIFTH ED.,1970.
C
      IF (REYN.LT.1000.) FF=.5/REYN
      IF (REYN.GE.1000.) FF=.0032149*(REYN**(-.2694))
C
C THE FRICTION FACTOR IS DIMENSIONAL, SQ.FT./SQ.IN.
C
C THE AREA BETWEEN REYN= 1000 TO 3000 IS UNSTABLE AND RESULTS ARE
C QUESTIONABLE IN THIS RANGE.
      RETURN
      END
```

```
      FUNCTION FLDEVS (T)
C
C GIVEN A TEMPERATURE IN DEGREES F, THE FUNCTION WILL CALCULATE THE
C DENSITY OF SATURATED WATER IN LBM/CU.FT.
C
      FLDENS=62.67137-.0024345*T-5.089E-5*T*T
C
      RETURN
      END
```

```
      SUBROUTINE LMTD (ITYPE,FLTMP1,FLTMP2,ARTMP1,ARTMP2,DELTM,FT,ALMTD,
     1P,Q,R,DIMIM)
C
      DIMENSION A(4,4)
C
C 'LMTD' CALCULATES MEAN TEMPERATURE DIFFERENCES FOR NINE COUNTERCUR-
C RENT CROSS-FLOW ARRANGEMENTS, AS APPLIED IN AIR COOLED HEAT EXCHANGERS
C THE CORRECTION FACTOR, FT, IS COMPUTED AND THEN APPLIED TO THE PURE
C COUNTERCURRENT FLOW LMTD.
C
C TYPE 1 - 1 ROW, 1 PASS
C TYPE 2 - 2 ROW, 1 PASS
C TYPE 3 - 3 ROW, 1 PASS
C TYPE 4 - 4 ROW , 1 PASS
C TYPE 5 - 2 ROW, 2 PASS
C TYPE 6 - 3 ROW,3 PASS
C TYPE 7 - 4 ROW,2 PASS
C TYPE 8 - 4 ROW, 4 PASS
C TYPE 9 - PURE CROSS FLOW
C TYPE 10 - PURE COUNTERFLOW
C FLTMP1 - TEMP OF FLUID ENTERING
C FLTMP2 - TEMP OF FLUID LEAVING
C ARTMP1 - TEMP OF AIR ENTERING
C ARTMP2 - TEMP OF AIR LEAVING
C CALCULATE LOG MEAN TEMPERATURE DIFF OF PURE COUNTERCURRENT
C              TEMP DIFF ACROSS SIDES OF EXCHANGER
C
      DEL1=(FLTMP1-ARTMP2)
C
      DEL2=(FLTMP2-ARTMP1)
C
C INSURE LN(DEL1/DEL2) DOES NOT GO TO ZERO.
C
      DUMMY0=DEL1/DEL2
C
      DUMMY1=FLTMP1-FLTMP2
C
      DUMMY2=ARTMP2-ARTMP1
C
      IF (DUMMY0.LT.1.01) GO TO 10
C
      ALMTD=(DEL1-DEL2)/ALOG(DUMMY0)
C
      GO TO 20
10    ALMTD=(FLTMP1/2.+FLTMP2/2.)-(ARTMP2/2.+ARTMP1/2.)
C
C INSURE DIMIM DOES NOT GO TO INFINITY.
C CALCULATE EFFECTIVENESS OF BOTH STREAMS.
```

```
C
20    P=DUMMY1/(FLTMP1-ARTMP1)
C
      Q=DUMMY2/(FLTMP1-ARTNP1)
C
      DUMMY3=(1.-Q)/(1.-P)
C
      IF (DUMMY3.LT.1.01) GO TO 30
C
C DIMENSIONLESS LMTD OF PURE COUNTERCURRENT.
C
      DIMIM=(P-Q)/(ALOG(DUMMY3))
C
      GO TO 40
C
30    DIMIM=ALMTD/(FLTMP1-ARTMP1)
C
C FOR PURE COUNTERFLOW
C
40    FT=1.
C
      IF (ITYPE.EQ.10) GO TO 160
C
C LISTING OF COEFFICIENTS FOR EACH TYPE ARRANGEMENT
C
      IF (ITYPE.GT.1) GO TO 50
      A(1,1)=-.462
      A(1,2)=5.08
      A(1,3)=-15.7
      A(1,4)=17.2
      A(2,1)=-.0313
      A(2,2)=.529
      A(2,3)=-2.37
      A(2,4)=3.18
      A(3,1)=-.174
      A(3,2)=1.32
      A(3,3)=-2.93
      A(3,4)=1.99
      A(4,1)=-.042
      A(4,2)=.347
      A(4,3)=-.853
      A(4,4)=.649
C
      GO TO 130
C
50    IF (ITYPE.GT.2) GO TO 60
C
      A(1,1)=-.334
```

```
      A(1,2)=3.3
      A(1,3)=-8.7
      A(1,4)=8.7
      A(2,1)=-.154
      A(2,2)=1.28
      A(2,3)=-3.35
      A(2,4)=2.83
      A(3,1)=-.0865
      A(3,2)=.546
      A(3,3)=-.929
      A(3,4)=.471
      A(4,1)=.0553
      A(4,2)=-.405
      A(4,3)=.953
      A(4,4)=-.717
C
      GO TO 130
C
60    IF (ITYPE.GT.3) GO TO 70
C
      A(1,1)=-.0874
      A(1,2)=1.05
      A(1,3)=-2.45
      A(1,4)=3.21
      A(2,1)=-.0318
      A(2,2)=.274
      A(2,3)=-.746
      A(2,4)=.668
      A(3,1)=-.0183
      A(3,2)=.123
      A(3,3)=-.156
      A(3,4)=.0617
      A(4,1)=.0071
      A(4,2)=-.0499
      A(4,3)=.109
      A(4,4)=-.0746
C
      GO TO 130
C
70    IF (ITYPE.GT.4) GO TO 80
C
      A(1,1)=-.0414
      A(1,2)=.615
      A(1,3)=-1.2
      A(1,4)=2.06
      A(2,1)=-.0139
      A(2,2)=.123
      A(2,3)=-.345
```

```
      A(2,4)=.318                                                       H1450
      A(3,1)=-.00723                                                    H1460
      A(3,2)=.0566                                                      H1470
      A(3,3)=-.0437                                                     H1480
      A(3,4)=.0111                                                      H1490
      A(4,1)=.0061                                                      H1500
      A(4,2)=-.0463                                                     H1510
      A(4,3)=.107                                                       H1520
      A(4,4)=-.0757                                                     H1530
C                                                                       H1540
      GO TO 130                                                         H1550
C                                                                       H1560
80    IF (ITYPE.GT.5) GO TO 90                                          H1570
C                                                                       H1580
      A(1,1)=-.235                                                      H1590
      A(1,2)=2.28                                                       H1600
      A(1,3)=-6.44                                                      H1610
      A(1,4)=6.24                                                       H1620
      A(2,1)=-.0773                                                     H1630
      A(2,2)=.632                                                       H1640
      A(2,3)=-1.63                                                      H1650
      A(2,4)=1.35                                                       H1660
      A(3,1)=-.0598                                                     H1670
      A(3,2)=.364                                                       H1680
      A(3,3)=-.613                                                      H1690
      A(3,4)=.276                                                       H1700
      A(4,1)=.00525                                                     H1710
      A(4,2)=-.0127                                                     H1720
      A(4,3)=-.0114                                                     H1730
      A(4,4)=.0272                                                      H1740
C                                                                       H1750
      GO TO 130                                                         H1760
C                                                                       H1770
90    IF (ITYPE.GT.6) GO TO 100                                         H1780
C                                                                       H1790
      A(1,1)=-.843                                                      H1800
      A(1,2)=5.85                                                       H1810
      A(1,3)=-12.8                                                      H1820
      A(1,4)=9.14                                                       H1830
      A(2,1)=.0302                                                      H1840
      A(2,2)=-.00964                                                    H1850
      A(2,3)=-.228                                                      H1860
      A(2,4)=.266                                                       H1870
      A(3,1)=.48                                                        H1880
      A(3,2)=-3.28                                                      H1890
      A(3,3)=7.11                                                       H1900
      A(3,4)=-4.9                                                       H1910
      A(4,1)=.0812                                                      H1920
```

```
      A(4,2)=-.834                                                      H1930
      A(4,3)=2.19                                                       H1940
      A(4,4)=-1.69                                                      H1950
C                                                                       H1960
      GO TO 130                                                         H1970
C                                                                       H1980
100   IF (ITYPE.GT.7) GO TO 110                                         H1990
C                                                                       H2000
      A(1,1)=-.605                                                      H2010
      A(1,2)=4.34                                                       H2020
      A(1,3)=-9.72                                                      H2030
      A(1,4)=7.54                                                       H2040
      A(2,1)=.0231                                                      H2050
      A(2,2)=.0059                                                      H2060
      A(2,3)=-.248                                                      H2070
      A(2,4)=.287                                                       H2080
      A(3,1)=.294                                                       H2090
      A(3,2)=-1.99                                                      H2100
      A(3,3)=4.32                                                       H2110
      A(3,4)=-3.                                                        H2120
      A(4,1)=.0198                                                      H2130
      A(4,2)=-.305                                                      H2140
      A(4,3)=.897                                                       H2150
      A(4,4)=-.731                                                      H2160
C                                                                       H2170
      GO TO 130                                                         H2180
C                                                                       H2190
110   IF (ITYPE.GT.8) GO TO 120                                         H2200
C                                                                       H2210
      A(1,1)=-.339                                                      H2220
      A(1,2)=2.38                                                       H2230
      A(1,3)=-5.26                                                      H2240
      A(1,4)=3.9                                                        H2250
      A(2,1)=.0277                                                      H2260
      A(2,2)=-.0999                                                     H2270
      A(2,3)=.0904                                                      H2280
      A(2,4)=-.000845                                                   H2290
      A(3,1)=.179                                                       H2300
      A(3,2)=-1.21                                                      H2310
      A(3,3)=2.62                                                       H2320
      A(3,4)=-1.81                                                      H2330
      A(4,1)=-.0199                                                     H2340
      A(4,2)=.04                                                        H2350
      A(4,3)=.0494                                                      H2360
      A(4,4)=-.0981                                                     H2370
C                                                                       H2380
      GO TO 130                                                         H2390
C                                                                       H2400
```

```
C  TYPE 9 - PURE CROSSFLOW
C
120    A(1,1)=.0669
       A(1,2)=-.278
       A(1,3)=1.11
       A(1,4)=.136
       A(2,1)=0.0
       A(2,2)=0.0
       A(2,3)=0.0
       A(2,4)=0.0
       A(3,1)=.0395
       A(3,2)=-.22
       A(3,3)=.454
       A(3,4)=-.258
       A(4,1)=0.0
       A(4,2)=0.0
       A(4,3)=0.0
       A(4,4)=0.0
C
C CALCULATION OF CORRECTION FACTOR, FT.
C
130    R=P/Q
C
       SUMI=0.0
C
       DO 150 I=1,4
C
       DO 140 K=1,4
C
       SUM=A(I,K)*((1.-DIMIM)**K)*SIN(2.*I*ATAN(R))
C
       SUMI=SUMI+SUM
C
140    CONTINUE
150    CONTINUE
C
       FT=1.-SUMI
C
C THE GIVEN TEMP DIFFERENCE FOR THE ARRANGEMENT ...
C
160    DELTM=FT*ALMTD
C
       RETURN
       END
```

```
      SUBROUTINE REFTEM (FLTMP1,FLTMP2,ARTMP1,ARTMP2,DELT1,DELT2,T1,T1P,
     1T2,T2P,FMDOT,CP,AMDOT,ACP,FLCAP,ARCAP)
C
      DELT1=((FLTMP1-ARTMP2)**.78868)*((FLTMP2-ARTMP1)**0.21132)
      DELT2=((FLTMP1-ARTMP2)**.21132)*((FLTMP2-ARTMP1)**0.78868)
C
C THESE ARE THE REFERENCE TEMPERATURE DIFFERENCES.
C
C CALCULATE UNCORRECTED REFERENCE TEMPERATURES.
C T1 - UNCORRECTED REFERENCE BULK TEMPERATURE OF TUBESIDE FLUID FOR
C        CALCULATING FIRST REFERENCE OVERALLL HEAT TRANSFER COEF.,UK1.
C T1P - UNCORRECTED REF. BULK TEMP. OF AIR FOR CALCULATING UK1.
C T2 - UNCORRECTED REF. BLK. TEMP. OF FLUID FOR CALC UK2.
C T2P - UNCORRECTED REF. BLK. TEMP. OF AIR FOR CALC UK2.
C
C CALCULATE CAPACITIES.
C
      FLCAP=FMDOT*CP
C
      ARCAP=AMDOT*ACP
C
C CALCULATE REFERENCE TEMPERATURES.
C
      DUMMY0=FLTMP2-ARTMP1
C
      DUMMY1=FLTMP1-ARTMP2
C
      IF (ABS(FLCAP-ARCAP).LT.100.) GO TO 10
C
      DUMMY2=(DELT1-DUMMY0)/(DUMMY1-DUMMY0)
C
      DUMMY3=(DELT2-DUMMY0)/(DUMMY1-DUMMY0)
C
      GO TO 20
C
10    DUMMY2=.78868
C
      DUMMY3=.21132
C
20    DUMMY4=FLTMP1-FLTMP2
C
      DUMMY5=ARTMP2-ARTMP1
C
      T1=FLTMP2+DUMMY4*DUMMY2
C
      T1P=ARTMP1+DUMMY5*DUMMY2
C
      T2=FLTMP2+DUMMY4*DUMMY3
```

```
C
      T2P=ARTMP1+DUMMY5*DUMMY3
C
      RETURN
      END
```

```
      SUBROUTINE CORECT (DELTM,ALMTD,T1,T1P,T2,T2P,SI1,SI2,SI1P,SI2P,CT,
     1CTP,FLCAP,ARCAP,DELT1,DELT2)
C
      DIMENSION CT(2),CTP(2)
C
C CALCULATE CORRECTIONS, SI, TO REFERENCE TEMPERATURES.
C
      DUMMY0=(1.-(DELTM/ALMTD))/(1.+((FLCAP/ARCAP)**0.66667))
C
      DUMMY1=(1.-(DELTM/ALMTD))/(1.+((ARCAP/FLCAP)**0.6667))
C
      SI1=DELT1*DUMMY0
C
      SI2=DELT2*DUMMY0
C
      SI1P=DELT1*DUMMY1
C
      SI2P=DELT2*DUMMY1
C
C THE CORRECTED REFERENCE BULK TEMPERATURES.
C
      CT(1)=T1-SI1
C
      CT(2)=T2-SI2
C
      CTP(1)=T1P+SI1P
C
      CTP(2)=T2P+SI2P
C
      RETURN
      END
```

```
      SUBROUTINE TUBES (BANKH,TUBEOD,FINHT,PITCHN,NROWS,NPASS,TUBEID,TAR
     1EA,TOTAL,IVROW,FLAREA,VROWR)
C
C VERTICAL ROWS CONTAIN (EQUAL NO. PER VERT. ROW) .......
C
      VROWR=(BANKH-(TUBEOD+2.*FINHT)-PITCHN/2.)/PITCHN
C
      IF (VROWR.LT.2.) VROWR=2.
C
C THIS IS A REAL NUMBER OF TUBES.
C
C TRUNCATING TO A WHOLE NUMBER OF TUBES.
      IVROW=VROWR
C
C TOTAL NUMBER OF TUBES IN THE BANK.
C
      TOTAL=IVROW*NROWS
C
C FLOW AREA PER TUBE IN SQ.IN.
C
      TAREA=3.14159*TUBEID*TUBEID/4.
C
C CALCULATE THE TOTAL FLOW AREA IN SQ.IN.
C
      IF (NPASS.EQ.1) GO TO 10
C
      IF (NPASS.EQ.NROWS) GO TO 20
C
      IF (NPASS.EQ.NROWS/2) FLAREA=TAREA*(2.*IVROW)
C
      GO TO 30
C
10    FLAREA=TAREA*TOTAL
C
      GO TO 30
C
20    FLAREA=TAREA*IVROW
C
30    RETURN
      END
```

```
      SUBROUTINE HTAREA (FINPIN,TUBEOD,FINHT,FINTH,AS,AT,ASAT,TOTAL,BANK
     1W)
C
C THIS SUBROUTINE CALCULATES THE AIRSIDE FIN AREA, THE TOTAL HEAT
C TRANSFER AREA AND THEIR RATIO.
C
C FINNED AREA PER TUBE PER INCH IN SQ.IN.
      DUMMY0=FINPIN*(((TUBEOD+2.*FINHT)**2-TUBEOD**2)*1.5708)
C
C BARE TUBE AREA PER INCH OF TUBE IN SQ.IN.
C
      DUMMY1=(1.-(FINPIN*FINTH))*TUBEOD*3.145927
C
C TOTAL INCHES OF TUBE IN BANK IN SQ.IN.
C
      DUMMY2=TOTAL*BANKW
C
C TOTAL FINNED AREA IN SQ. IN.
C
      AS=DUMMY0*DUMMY2
C
C TOTAL HEAT TRANSFER AREA IN SQ.IN.
C
      AT=(DUMMY0+DUMMY1)*DUMMY2
C
C RATIO OF FINNED AREA TO TOTAL
C
      ASAT=AS/AT
C
      RETURN
      END
      SUBROUTINE FILMI (CT,TUBEID,FMDOT,FLAREA,L,CP,BANKW,HI)
C
C ***THIS SUBROUTINE IS FOR FORCED CONVECTION OF   SINGLE PHASE,
C INCOMPRESSIBLE FLUID IN A SMOOTH TUBE.
C
      DIMENSION CT(2),HI(2)
C
      DO 10 I=1,2
C
C THE REYNOLDS NO.   ...............
C
C VISCOS - VISCOSITY OF THE FLUID
      VISCOS=VISCFL(CT(I))
C
      REYN=(TUBEID*FMDOT*12.)/(FLAREA*VISCOS)
C
C IS THE FLOW TURBULENT, LAMINAR OR IN TRANSITION?
```

```
C
      IF (REYN.GT.10000.) L=0
      IF (REYN.LT.2100.) L=1
      IF (REYN.GE.2100.0.AND.REYN.LE.10000.) L=2
C
C TC - THERMAL CONDUCTIVITY OF THE FLUID.
C
      TC=TCFL(CT(I))
C
C THE PRANDLT NO.  .........
      PR=(CP*VISCOS)/TC
C
C THE NUSSELT NO.  .......
C
C ... SEIDER - TATE CORRELATION ............
      IF (L.EQ.1) FLNUS=1.86*((REYN*PR)**.333333)*((TUBEID/BANKW)**.3333
     133)
C
      IF (L.EQ.0) FLNUS=.027*(REYN**.8)*(PR**.333333)
C
C ... HAUSEN CORRELATION ...
C
      IF (L.EQ.2) FLNUS=.116*((REYN**.666667)-125.)*(PR**.333333)*(1.+((
     1TUBEID/BANKW)**.666667))
C
C THE FILM COEFFICIENT .......
C
      HI(I)=(FLNUS*TC*12.)/TUBEID
C
10    CONTINUE
C
      RETURN
      END
```

```
      SUBROUTINE VMAX (ARTMP1,PRES1,GASCON,AMDOT,BANKH,BANKW,FINSP,FINTH
     1,FINHT,TUBEOD,IVROW,STOTAL,SFF,FINPIN,SFIN,SROOT,VMAXF,VMAXS,DELSF
     2F)
C
C CALCULATE DENSITY AT INLET CONDITIONS IN   LBM/CU.FT.....
C CONVERT TO ABS TEMPERATURE.
C
      ARTMPI=ARTMP1+460.
C
      RHO=(PRES1*144.)/(GASCON*ARTMPI)
C
C PRESSURE WAS CONVERTED TO PSF.
C
      DO 10 I=1,2
C
C CALCULATE MINIMUM FLOW AREA IN SQ.IN.   .......
C NUMBER OF FINS PER INCH.
C
      FINPIN=1./FINSP
C
C PROJECTED FIN AREA PER TUBE IN SQ.IN.
C
      SFIN=FINTH*2.*FINHT*FINPIN*BANKW
C
C PROJECTED ROOT TUBE AREA PER TUBE IN SQ.IN.
C
      SROOT=TUBEOD*BANKW
C
C TOTAL PROJECTED AREA IN SQ.IN.
C
      STOTAL=IVROW*(SFIN+SROOT)
C
      DELSFF=STOTAL-BANKH*BANKW
C
C MAXIMUM VELOCITY AT FREE STREAM CONDITIONS IN FT/MIN .....
C THE FREE FACE AREA IN SQ.IN.
C
      SFF=(BANKH*BANKW)-STOTAL
C
      IF (SFF.LT..001) SFF=1.
C
      VMAXS=(144.*AMDOT)/(60.*RHO*SFF)
C
      IF (I.EQ.1) VMAXF=VMAXS
C
      IF (I.EQ.2) RETURN
C
C NOW CALCULATE VMAX AT STND CONDITIONS FOR CORRECTION OF AIR FILM
```

```
C COEFFICIENT.
C
      RHO=.074
C
10    CONTINUE
C
      END
```

```
      SUBROUTINE FILM3 (TUBEOD,AMDOT,SFF,FINSP,FINTH,S,FINHT,CTP,HO,ACP)
C
C THE UNCORRECTED AIRSIDE FILM COEFFICIENT CAN BE CALCULATED DIRECTLY
C USING THE BRIGGS-YOUNG CORRELATION. THE CORRELATION WILL YIELD GOOD
C RESULTS FOR TRIANGULAR PITCH BANKS OF HIGH FINNED TUBES WITH SIX
C ROWS. FOR OTHER THAN SIX ROWS A CORRECTION IS NECESSARY.
C
      DIMENSION CTP(2),HO(2)
C
      DO 10 I=1,2
C
C FIRST, THE REYNOLDS NO.     .........
C ... VISCOSITY ...
C
      VISCA=VISCAR(CTP(I))
C
      REYNA=(TUBEOD*AMDOT*12.)/(SFF*VISCA)
C
C THE PRANDLT NO.  ..................
C ... THERMAL CONDUCTIVITY ...
C
      TCA=TCAR(CTP(I))
C
      PRA=(ACP*VISCA)/TCA
C
C CALCULATE THE DISTANCE BETWEEN FINS IN INCHES .........
C
      S=FINSP-FINTH
C
C CALCULATE THE NUSSELT NO.  .................
C
      ARNUS=.1378*(REYNA**.718)*(PRA**.333333333)*((S/FINHT)**.296)
C
C THE FILM COEFFICIENT IN BTU/HR-SQ.FT-F  ..............
C
      HO(I)=(ARNUS*TCA*12.)/TUBEOD
C
10    CONTINUE
C
      RETURN
      END
```

```
      SUBROUTINE ROWCOR (VMAXS,NROWS,C)
C
C ....CORRECT OUTSIDE FILM COEFFICIENT FOR NUMBER OF ROWS ......
C
C CORRECTIONS ARE FROM A GRAPH BY WARD AND YOUNG FOR NUMBER OF ROWS
C IN A TUBE BANK AS COMPARED TO SIX.
      IF (NROWS.GT.1) GO TO 10
C
C  INTERPOLATION .....
C
      IF (VMAXS.GE.300..AND.VMAXS.LT.500.) C=(.8505-(((VMAXS-300.)*.0405
     1)/200.))/(.975-(((VMAXS-300.)*.014)/200.))
C
      IF (VMAXS.GE.500..AND.VMAXS.LE.1000.) C=(.81-(((VMAXS-500.)*.065)/
     1500.))/(.961-(((VMAXS-500.)*.011)/500.))
C
C  **CAUTION**  EXTRAPOLATION ...
C
      IF (VMAXS.LT.300.) C=(.8505+(((300.-VMAXS)*.1055)/700.))/(.975+(((
     1300.-VMAXS)*.025)/700.))
C
      IF (VMAXS.GT.1000.) C=(.745-(((VMAXS-1000.)*.1055)/700.))/(.95-(((
     1VMAXS-1000.)*.025)/700.))
C
      GO TO 50
C
10    IF (NROWS.GT.2) GO TO 20
C
C   INTERPOLATION .........
C
      IF (VMAXS.GE.300..AND.VMAXS.LT.500.) C=(.92-(((VMAXS-300.)*.04)/20
     10.))/(.975-(((VMAXS-300.)*.014)/200.))
C
      IF (VMAXS.GE.500..AND.VMAXS.LE.1000.) C=(.88-(((VMAXS-500.)*.05)/5
     100.))/(.961-(((VMAXS-500.)*.011)/500.))
C
C  **CAUTION**  EXTRAPOLATION ...........
C
      IF (VMAXS.LT.300.) C=(.92+(((300.-VMAXS)*.09)/700.))/(.975+(((300.
     1-VMAXS)*.025)/700.))
C
      IF (VMAXS.GT.1000.) C=(.83-(((VMAXS-1000.)*.09)/700.))/(.95-(((VMA
     1XS-1000.)*.025)/700.))
C
      GO TO 50
C
20    IF (NROWS.GT.3) GO TO 30
C
```

```
C     INTERPOLATION ..........
C
       IF (VMAXS.GE.300..AND.VMAXS.LT.500.) C=(.95-(((VMAXS-300.)*.032)/2
     100.))/(.975-(((VMAXS-300.)*.014)/200.))
C
       IF (VMAXS.GE.500..AND.VMAXS.LE.1000.) C=(.918-(((VMAXS-500.)*.033)
     1/500.))/(.961-(((VMAXS-500.)*.011)/500.))
C
C   **CAUTION**  EXTRAPOLATION ......
C
       IF (VMAXS.LT.300.) C=(.95+(((300.-VMAXS)*.065)/700.))/(.95+(((300.
     1-VMAXS)*.025)/700.))
C
       IF (VMAXS.GT.1000.) C=(.885-(((VMAXS-1000.)*.065)/700.))/(.95-(((V
     1MAXS-1000.)*.025)/700.))
C
       GO TO 50
C
30     IF (NROWS.GT.4) GO TO 40
C
       IF (VMAXS.GE.300..AND.VMAXS.LT.500.) C=(.965-(((VMAXS-300.)*.025)/
     1200.))/(.975-(((VMAXS-300.)*.014)/200.))
C
       IF (VMAXS.GE.500..AND.VMAXS.LE.1000.) C=(.94-(((VMAXS-500.)*.025)/
     1500.))/(.961-(((VMAXS-500.)*.011)/500.))
C
C THIS WAS THE INTERPOLATION FOR THE CORRECTION FACTOR FOR FOUR ROWS.
C
C **CAUTION**  EXTRAPOLATION ....
       IF (VMAXS.LT.300.) C=(.965+(((300.-VMAXS)*.05)/700.))/(.975+(((300
     1.-VMAXS)*.025)/700.))
C
       IF (VMAXS.GT.1000.) C=(.915-(((VMAXS-1000.)*.05)/700.))/(.95-(((VM
     1AXS-1000.)*.025)/700.))
C
       GO TO 50
C
40     ROWS=NROWS
C
       IF (VMAXS.GE.300..AND.VMAXS.LT.500.) C=((.02175*ALOG(ROWS)+.9349)-
     1(((VMAXS-300.)*((.02175*ALOG(ROWS)+.9349)-(.036*ALOG(ROWS)+.892)))
     2/200.))/(.975-(((VMAXS-300.)*.014)/200.))
C
       IF (VMAXS.GE.500..AND.VMAXS.LE.1000.) C=((.036*ALOG(ROWS)+.892)-((
     1(VMAXS-500.)*((.036*ALOG(ROWS)+.892)-(.036*ALOG(ROWS)+.877)))/500.
     2))/(.961-(((VMAXS-500.)*.011)/500.))
C
C    *** CAUTION EXTRAPOLATION
```

```
C
      IF (VMAXS.LT.300.) C=((.02175*ALOG(ROWS)+.9349)+(((300.-VMAXS)*((.
     102175*ALOG(ROWS)+.9349)-(.036*ALOG(ROWS)+.877)))/700.))/(.975+(((3
     200.-VMAXS)*.025)/700.))
C
      IF (VMAXS.GT.1000.) C=((.036*ALOG(ROWS)+.877)-(((VMAXS-1000.)*((.0
     12175*ALOG(ROWS)+.9349)-(.036*ALOG(ROWS)+.877)))/700.))/(.95-(((VMA
     2XS-1000.)*.025)/700.))
C
50    RETURN
      END
```

```
      SUBROUTINE FINEFF (FINHT,TUBEOD,AK,FINTH,HO,JTYPE,EATA,AM,RO,RE,BE
     1TA,ARG1,ARG2,B2,B1)
C
C 'FINEFF' CALCULATES THE FIN EFFICIENCY OF VARIOUS PROFILES USING
C SIMPLIFIED CONSTRAINTS.
C
C BETA - FIN THICKNESS, FEET
C RE - RADIUS TO EDGE OF FIN, FEET
C RO - RADIUS TO BASE OF FIN, FEET
C HO - FILM COEFFICIENT, BTU/SQ.FT-HR-F
C AK - THERMAL CONDUCTIVITY OF FIN MATERIAL, BTU/FT-HR-F
C TYPE 1 - RADIAL.... RECTANGULAR PROFILE
      DIMENSION HO(2),EATA(2)
C
      DO 10 I=1,2
C
      IF (JTYPE.GT.1) GO TO 20
C
C --IN FEET--
C
      RO=TUBEOD/24.
C
      RE=RO+FINHT/12.
C
      BETA=FINTH/12.
C
      AM=SQRT((2.0*HO(I))/(AK*BETA))
C
      ARG1=AM*RE
C
      ARG2=AM*RO
C
      B2=BK1(ARG1)
C
      B1=BI1(ARG1)
C
      EATA(I)=((2.0*RO)/(AM*(RE*RE-RO*RO)))*((B1*BK1(ARG2)-B2*BI1(ARG2))
     1/(BI0(ARG2)*B2+B1*BK0(ARG2)))
C
10    CONTINUE
C
20    GO TO 30
30    RETURN
      END
```

```
      FUNCTION BIO (X)
C
C     TAKEN FROM REFERENCE (46).
C
      T=X/3.75
C
      IF (ABS(X)-3.75) 10,10,20
C
10    BIO=1.+3.5156229*T**2+3.0899424*T**4+1.2067492*T**6+0.2659732*T**8
     1+0.0360768*T**10+0.0045813*T**12
C
      RETURN
C
20    BIO=(0.39894228+0.01328592/T+0.00225319/T**2-0.00157565/T**3+.0091
     16231/T**4-.02057706/T**5+.02635537/T**6-.01647633/T**7+.00392377/T
     2**8)*EXP(X)/SQRT(X)
C
      RETURN
      END
```

```
      FUNCTION BI1 (X)
C
C     TAKEN FROM REFERENCE (46).
C
      T=X/3.75
C
      IF (ABS(X)-3.75) 10,10,20
C
10    BI1=(.5+.87890594*T**2+.51498869*T**4+.15084934*T**6+.02658733*T**
     18+.00301532*T**10+.00032411*T**12)*X
C
      RETURN
C
20    BI1=(.39894228-.03988024/T-.00362018/T**2+.00163801/T**3-.01031555
     1/T**4+.02282967/T**5-.02895312/T**6+.01787654/T**7-.00420059/T**8)
     2*EXP(X)/SQRT(X)
C
      RETURN
      END
```

```
      FUNCTION BKO (X)
C
C    TAKEN FROM REFERENCE (46).
C
      T=X/2
C
      IF (X-2.) 10,10,20
C
10    BKO=-ALOG(T)*BIO(X)-.57721566+.4227842*T**2+.23069756*T**4+.034885
     19*T**6+.00262698*T**8+.0001075*T**10+.0000074*T**12
C
      RETURN
C
20    BKO=(1.25331414-.07832358/T+.02189568/T**2-.01062446/T**3+.0058787
     12/T**4-.0025154/T**5+.00053208/T**6)/SQRT(X)/EXP(X)
C
      RETURN
      END
```

```
      FUNCTION BK1 (X)
C
C     TAKEN FROM REFERENCE (46).
C
      T=X/2
C
      IF (X-2.) 10,10,20
C
10    BK1=ALOG(T)*BI1(X)+(1.+.15443144*T**2-.67278579*T**4-.18156897*T**
     16-.01919402*T**8-.00110404*T**10-.00004686*T**12)/X
C
      RETURN
C
20    BK1=(1.25331414+.23498619/T-.0365562/T**2+.01504268/T**3-.00780353
     1/T**4+.00325614/T**5-.00068245/T**6)/SQRT(X)/EXP(X)
C
      RETURN
      END
```

```
      SUBROUTINE CAY (HI,W,TUBEID,RESAIR,TUBEOD,CT,CTP,CA)
      DIMENSION HI(2),RESAIR(2),CT(2),CTP(2),CA(2)
C
C USING AN ITERATIVE-FREE METHOD OF ROETZEL, CALCULATE THE CORRECTION
C FACTOR TO THE TUBESIDE FILM COEFFICIENT.
C
      DO 10 I=1,2
C
C THE RATIO OF THE UNCORRECTED HEAT TRANSFER RESISTANCES. TUBEID HAD TO
C BE CONVERTED TO A RADIUS IN FT.
C
      AI=HI(I)*(W*(TUBEID/24.)+RESAIR(I)*(TUBEID/TUBEOD))
C
C GUESS A WALL TEMPERATURE.
C
      WLTMP=(CT(I)+CTP(I))/2.
C
C CALCULATE THE VISCOSITY AT THE REFERENCE BULK TEMPERATURE,CT(I), AND
C AT THE ESTIMATED WALL TEMPERATURE AND FORM THEIR RATIO.
C
      VISCOS=VISCFL(CT(I))
C
      VISCPR=VISCFL(WLTMP)
C
      DUMMYO=VISCPR/VISCOS
C
C CALCULATE U AND VV IN ACCORDANCE WITH ROETZEL'S EXPRESSIONS.
C
      U=.07*ALOG(DUMMYO)*(1./AI)*((1.-((CTP(I)+460.)/(CT(I)+460.)))/(((C
     1T(I)+460.)/(WLTMP+460.))-1.))+(CTP(I)+460.)/((CT(I)+460.)*AI)-1.
C
      VV=-.07*ALOG(DUMMYO)*(1./AI)*((1.-((CTP(I)+460.)/(CT(I)+460.)))/((
     1(CT(I)+460.)/(WLTMP+460.))-1.))+(CTP(I)+460.)/((CT(I)+460.)*AI)
C
C THE CORRECTION, CA, IS:
C
      CA(I)=-U/2.+SQRT((U*U)/4.+VV)
C
10    CONTINUE
C
      RETURN
      END
```

```
      SUBROUTINE DELTAP (CTP,TUBEOD,AMDOT,SFF,PITCHN,PITCHL,PRES1,GASCON
     1,NROWS,UK,DELPA,CDELPA,SVOL,ARG5)
C
      DIMENSION CTP(2),DELPAR(2),UK(2),DUMMY(2)
C
C THIS SUBROUTINE CALCULATES THE AIRSIDE PRESSURE DROP, DELPA, USING
C THE ROBINSON - BRIGGS CORRELATION FOR TRIANGULAR PITCH BANKS OF HIGH-
C FINNED TUBES.
C
C ROETZEL'S METHOD WHICH TAKES INTO ACCOUNT THE AIR'S DEPENDENCE ON
C DENSITY AND ITS CHANGING PROPERTIES THROUGH THE TUBE BANK WILL BE
C INCORPORATED TO CORRECT DELPA .
C THE REFERENCE PRESSURE DROPS WILL BE CALCULATED.
      DO 10 I=1,2
C
C CALCULATE THE VISCOSITY OF AIR AT THE REFERENCE TEMPERATURE.
C
      VISCA=VISCAR(CTP(I))
C
C THE REYNOLDS NO. ........
C
      REYNA=(TUBEOD*AMDOT*12.)/(SFF*VISCA)
C
C CALCULATE THE FRICTION FACTOR ... FR. NOTE ITS LACK OF DEPENDENCE
C ON THE NUMBER OF ROWS.
C
      FR=18.93*(REYNA**(-.316))*((PITCHN/TUBEOD)**(-.927))*((PITCHN/PITC
     1HL)**.515)
C
C BEFORE THE PRESSURE DROP CAN BE CALCULATED THE REFERENCE DENSITY MUST
C BE CALCULATED FOR THE INLET PRESSURE IN LBM/CU.FT.
C
      RHO=(PRES1*144.)/(GASCON*(CTP(I)+460.))
C
      DUMMY(I)=RHO
C
C THE PRESSURE DROP IS CALCULATED IN PSI .....
C
      DELPAR(I)=(FR*NROWS*AMDOT*AMDOT*144.)/(4.18E8*SFF*SFF*RHO)
C
C WHERE 4.18E 8 IS A CONSTANT IN FT/SQ.HR. ; SFF, THE FREE FACE AREA IS
C IN SQ. IN.
C
10    CONTINUE
C
C FROM ROETZEL'S EQ. (47), THE UNCORRECTED PRESSURE DROP IS .....
C
      DELPA=((DELPAR(1)/UK(1))+(DELPAR(2)/UK(2)))/((1./UK(1))+(1./UK(2))
```

```
     1)
C
C AS A CONTROL DURING OPTIMIZATION; TO ENSURE THAT THE SQRT OF A
C NEGATIVE NJMBER IS NOT TAKEN DURING THE FINAL CORRECTION:
C
      ARG5=1.-(2.*DELPA/PRES1)
C
      CDELPA=DELPA
C
      IF (ARG5.GT.0.0) CDELPA=PRES1*(1.-SQRT(ARG5))
C
C WHERE CDELPA IS THE FINAL CORRECTED PRESSURE DROP.
C
C THE MEAN SPECIFIC VOLUME OF THE AIR FLOWING THROUGH THE EXCHANGER
C IN CU.FT./IBM.
      SVOL=2./(DUMMY(1)+DUMMY(2))
C
      RETURN
      END
```

```
      SUBROUTINE DELP (CT,TUBEID,FMDOT,FLAREA,CA,UK,DELPW,NPASS,BANKW)
C
      DIMENSION CT(2),CA(2),DP(2),UK(2)
C
C THE TUBESIDE PRESSURE DROP IS CALCULATED IN ACCORDANCE WITH THE
C STANDARDS OF THE TUBULAR  EXCHANGER MANUFACTURER'S ASSOCIATION.
C
C ROETZEL'S CORRECTION FOR CHANGING FLUID PROPERTIES IS APPLIED.
      DO 10 I=1,2
C
C THE VISCOSITY AT THE REFERENCE BULK TEMPERATURE.
C
      VISCOS=VISCFL(CT(I))
C
C THE REYNOLDS NO. ......
C
      REYN=(TUBEID*FMDOT*12.)/(FLAREA*VISCOS)
C
C THE TUBESIDE FRICTION FACTOR.
C
      F=FF(REYN)
C
C ARECCOMMMENDED CORRECTION FACTOR :
C
      IF (REYN.GT.2100.) PHI=CA(I)
      IF (REYN.LE.2100.) PHI=CA(I)**1.7857
C
C THE DENSITY AT THE REFERENCE TEMPERATURE IS CALCULATED IN LBM/CU.FT.
C
      RHO=FLDENS(CT(I))
C
C CALCULATE THE PRESSURE DROP DISREGARDING EXIT AND ENTRANCE LOSSES.
C
      DEP=(F*FMDOT*FMDOT*BANKW*NPASS*20736.)/(2.*4.17E8*RHO*TUBEID*PHI*F
     1LAREA*FLAREA)
C
C TO THIS ADD THE ADDITIONAL LOSSES. ( EQ.(9-11) KERN & KRAUS)
C
      ADD=((NPASS-1)*FMDOT*FMDOT*144.)/(2.*RHO*FLAREA*FLAREA*32.2*1.296E
     17)
C
C TO GET THE TOTAL PRESSURE DROP ....
C
      DP(I)=DEP+ADD
C
10    CONTINUE
C
C THE ACTUAL TUBESIDE PRESSURE DROP IN PSI .........
```

```
C
      DELPH=((DP(1)/UK(1))+(DP(2)/UK(2)))/((1./UK(1))+(1./UK(2)))
C
      RETURN
      END
```

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